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# TWO STAGE SIBLING CYCLE COMPRESSOR/EXPANDER


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**February 1994**



## Final Report

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14. Abstract

A two stage Sibling Cycle Refrigerator was investigated. A single stage refrigerator existed, and two models were used to design a second stage. The second stage was added to the existing cooler and tested (different combinations of port timing, pressure, and operating speed were considered). The test results were compared to models. Following testing, an investigation was conducted to try to explain differences between the analytical and experimental results. Finally, available machinery to drive devices which both rotate and reciprocate were reviewed.

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# PREFACE

Successful completion of this Phase I project was made possible by the imaginative cooperation of many people. Ran Yaron contributed many important suggestions for apparatus design and furnished his Stirling cycle code to model the piston/displacer motions that are unique to the Sibling Cycle. Robert A. Bilbrey contributed many valuable suggestions for detailed design of the modified second stage assembly that was at the heart of the experiments performed under the contract. Drazen Fabris set up the data acquisition system and participated extensively in the experimental work. He also participated actively in computer modelling of second stage performance. Not least were the contributions of highly skilled artisans, including Jim Olson, who executed delicate welds precisely and on very short notice.

The assistance of R. Warren Breckenridge and Thomas P. Hosmer of Arthur D. Little, Inc. in analyzing the suitability of a surplus rotating/reciprocating drive system and in analyzing gas bearings is gratefully acknowledged. Reuven Unger of Sunpower, Inc., supplied information on an electromagnetic linear drive system and provided helpful suggestions for drive system design.

Particular recognition is also due to: Capt. Pete Thomas, USAF, technical monitor on the project, whose suggestions and assistance were invaluable; Floyd Martinez, contracting officer, who smoothed the contracting process, and Rudy Chavez, who handled the details of contract negotiation and several subsequent modifications cheerfully and efficiently. Dorothy Sandoval did an incisive editing job on the manuscript.

Finally, thanks are due to Carl Nelson of BMDO, whose continuing support of research on the Sibling Cycle has given a major boost to development of this new technology.

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# 1.0 INTRODUCTION

Mitchell/Stirling Machines/Systems, Inc. (MS*2) experimented with a two-stage Sibling Cycle cryocooler. The Sibling's potential advantages are mechanical simplicity (one moving part) and light weight with projected performance at least comparable to that of Stirling Cycle coolers. The test apparatus was the first two-stage Sibling Cycle machine ever built.

The Phase I effort involved four main components:

1. Theoretical analysis with computer models;

2. Design and construction of a second stage to fit MS*2's existing Sibling Cycle refrigerator;

3. Experimental work with the two-stage Sibling Cycle hardware; and

4. Survey work to identify a feasible rotating/reciprocating electromagnetic drive to be used in Phase II.

All components of the contract were completed. However, the experimental results were well below those predicted by the computer models. Various changes were made in the hardware, without demonstrable improvement in results. Analysis of the results has identified major problems with the existing design and charted the way to better performance in the next attempt.

Several drive systems were considered as substitutes for the existing mechanical drive and design criteria for a linear/rotating electromagnetic drive were investigated and identified.

Successful demonstration of a cryocooler based upon the two-stage Sibling Cycle concept will require additional work on the first stage compressor/expander as well as an improved drive system in Phase II.

# 2.0 THE BASICS OF THE SIBLING CYCLE AND TEST RIG

The two-stage refrigerator used in this project combined two first-of-its-kind elements. The first stage hardware used in this contract was the first ported-piston Sibling Cycle compressor/expander ever built. The second stage hardware was the first second stage ever fitted to a Sibling Cycle first stage.

The first stage was built as a high-temperature, low-lift refrigerator, not as a cryocooler. The first stage unit was completed shortly before this contract was awarded, and testing of the original single-stage configuration was suspended so that the unit could be used in the performance of this contract.

The second stage was designed, built and fitted to the first stage as part of the performance of this contract. The intent was to use the pressure wave generated in the first stage to create a refrigeration effect in the expansion space of the second stage.

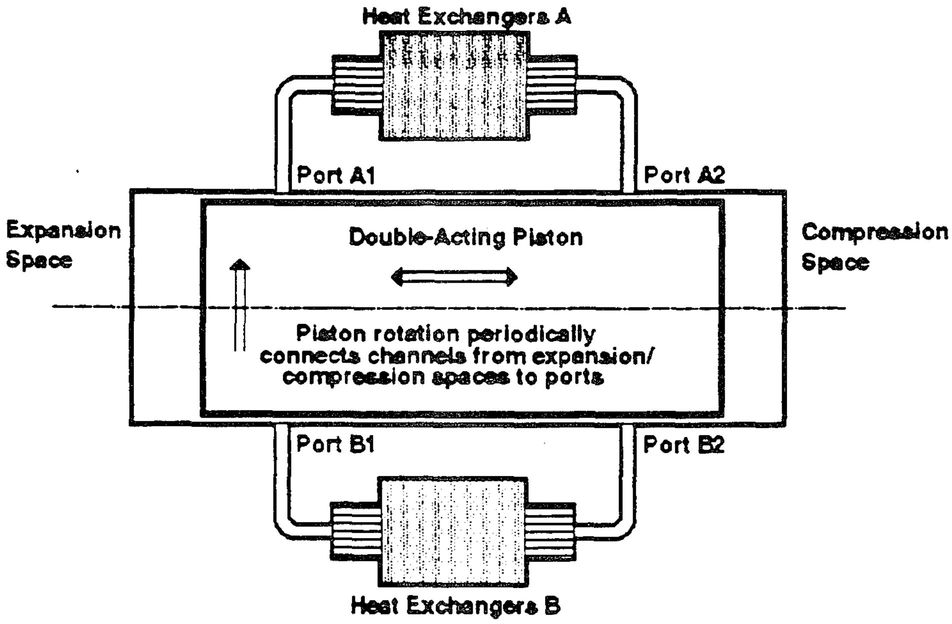
The first stage generates a pressure/volume relationship that is unique to the Sibling Cycle. The Sibling Cycle resembles both the Stirling cycle and the Ericsson cycle. It is, however, subtly different. While piston motion is continuous (essentially sinusoidal), fluid motion is discontinuous. Two separate masses of fluid undergo compression, expansion and transfer sequentially, using the same expansion and compression spaces. The cycle and the machine are covered by U.S. and foreign patents; the United States has certain rights in the second patent, which covers improvements conceived during a previous SBIR contract (Refs. 1-3) The principle and peculiarities of the machine and of its cycle have been described in published papers. (Refs. 4-9).

## 2.1 THE SIBLING CYCLE PRINCIPLE

The basic concept of the Sibling Cycle is illustrated in Figure 2.1.

MS*2's first stage Sibling refrigerator consists of a ported, double-acting piston/displacer simultaneously rotating and reciprocating in a ported cylinder. As the piston reciprocates, its rotation alternately opens and closes ports in the cylinder walls, controlling flows between the spaces swept by the two ends of the piston. The porting sequence is arranged so that compression takes place in one end of the cylinder and expansion in the other end.

The cylinder ports are normally connected axially through heat exchanger assemblies, each consisting of a freezer, regenerator, and cooler. At top dead center, with the piston all the way to the left in Figure 2.1, piston ports align themselves with cylinder ports, permitting one mass of fluid to enter the expansion end of the cylinder. As the piston moves to the right, away from top dead center, that fluid cools as it expands out of a set of heat exchangers into the expansion end of the cylinder.




| Piston moves | Ports | | | | Process | |
|---|---|---|---|---|---|---|
| | A1 | A2 | B1 | B2 | Side A | Side B |
| Left | Closed | Open | Open | Closed | Compression | Expansion |
| Right | Closed | Closed | Open | Open | - | Transfer |
| Left | Open | Closed | Closed | Open | Expansion | Compression |
| Right | Open | Open | Closed | Closed | Transfer | - |

Figure 2.1. Sibling Cycle porting sequence.

At the same time, the other end of the piston forces a second mass of fluid out of the compression space through the warm end of a different set of heat exchangers. It is important to note that the two piston ends are acting on separate fluid volumes at this stage in the cycle.

At the end of the stroke (bottom dead center) the rotation of the piston/displacer closes off ports at the compression end of the cylinder, trapping compressed fluid in a set of heat exchangers at maximum pressure. Before this compressed fluid can be released, the piston/displacer must return to top dead center, forcing fluid in the expansion space through a set of heat exchangers into the compression space.

In the simplest conception of the Sibling Cycle, the transfer stroke begins at bottom dead center and ends at top dead center with ports open at both ends of, alternately, the "A" side or "B" side heat exchangers to allow fluid to pass from the expansion space to the compression space. That idealized view would make sense if the piston swept the entire expansion space and the entire compression space and the ports opened and closed instantly and completely. There would be no residue of low pressure fluid in the expansion space

when the high pressure ports opened at top dead center. There would be no residue of high pressure fluid in the compression space when the low pressure ports opened at the compression end at bottom dead center.

Reality, however, is somewhat different. Ducts and clearances create unavoidable dead volumes in both expansion and compression spaces. Thus, to avoid an inrush of high pressure fluid at the expansion end at top dead center, all ports at the expansion end are closed somewhat before top dead center in order to pre-compress the fluid in the dead volume there. Then, when the high pressure ports open at the expansion end, pressure is essentially the same in the expansion end of the cylinder as in the high pressure heat exchanger assembly and there is no inrush of high pressure fluid.

Similarly, at the compression end, the opening of the low pressure ports is delayed until somewhat after the piston passes bottom dead center. As the piston moves away from bottom dead center, all ports remain closed for a time. Pressure in the dead volume drops. When the low pressure ports open, the pressure in the compression end of the cylinder is essentially the same as the pressure behind the newly-opened ports and there is no outrush of fluid from the compression space.

As the test apparatus was configured for this project, the compression end port did not open until 37 degrees past bottom dead center. At the other end of the transfer stroke, the port at the expansion end of the cylinder closed 30 degrees before top dead center. ("Degrees" are relative to piston reciprocation, not rotation; there are 360 degrees of crank turn in one reciprocation.)

To balance the forces on the piston, each set of heat exchangers was split into two parts, diametrically opposed. The cylinder had four ports at each end, and the piston two. The piston ports were elongated along a sinusoidal path, permitting them to cover the cylinder port position as the piston simultaneously rotated and reciprocated. Piston motion was controlled by a gearbox that delivered one complete rotation of the piston for every four reciprocations.

## 2.2 THERMODYNAMICS OF THE SIBLING CYCLE

The thermodynamic cycle of the first stage is reflected in the schematic expansion space pV diagram in Figure 2.2. From A to B is expansion, in which fluid flows into the expansion space from a set of heat exchangers; the port at the compression end of that set of heat exchangers is closed. During the entire piston stroke, pressure continuously decreases and fluid temperature falls.

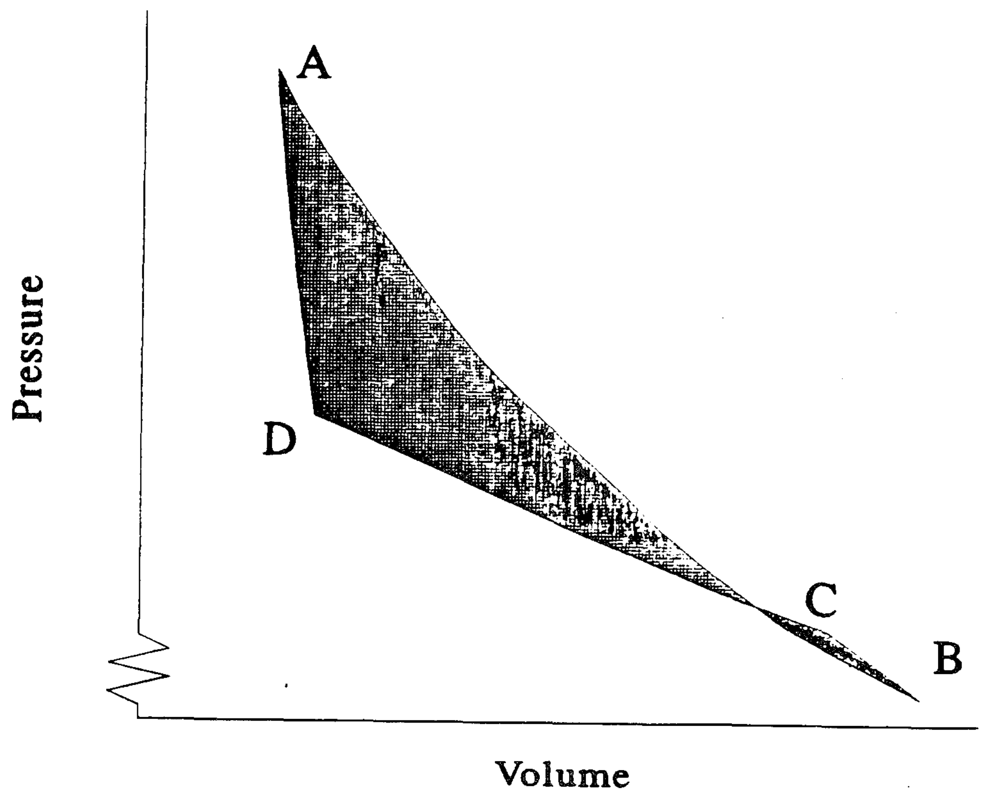



Figure 2.2. The pV diagram, expansion space of a Sibling Cycle refrigerator.

From B to A (through C and D), the piston returns. That is the "exchange" stroke; during most of the stroke (i.e., from C to D), ports are open at both ends of a set of heat exchangers and fluid is transferred to the compression end of the cylinder. There is little change in pressure during the exchange stroke if the temperature difference between expansion and compression spaces is small. If the temperature difference is large, the slope of the bottom line of the pV diagram can be controlled by using a stepped piston, with smaller bore at the expansion end.

The portions of the pV diagram from B to C and from D to A show the effects of timing the opening and closing of ports. From B to C, fluid is leaving the expansion space, but the port at the compression end of the heat exchangers is not yet open. The cycle is simply reversed, with fluid returning into a set of heat exchangers exactly as it left. Pressure rises because fluid cannot yet escape into the compression space.

From D to A, pressure rises rapidly. That is because the ports at the expansion end close before the piston reaches the end of its travel, trapping fluid in the expansion space. The

purpose of the early port closing is to drive up pressure in the expansion space to the same level as the pressure of the fluid compressed in the set of heat exchangers that will next discharge fluid into the expansion space.

Refrigeration is reflected by the large area of the pV diagram. Irreversible heat transfer, represented by the small area of the pV diagram, must be deducted to arrive at the net pV refrigeration. Other losses not modelled, such as conduction, will also affect the result. A positive pV diagram is a necessary condition for refrigeration, but not alone sufficient to insure cooling.

## 2.3 THE TWO-STAGE CONCEPT

The second stage of a two-stage Sibling refrigerator is different from the first stage in that it does not have ports. As in a Stirling Cycle cooler, the passage from the second stage expansion space through the second stage freezer tube and the second stage regenerator is open throughout its length at all times. Fluid flows freely back and forth in the second stage in response to pressure changes that are generated in part by the first stage and in part by the motion of the second stage piston. The concept is illustrated schematically in Figure 2.3.

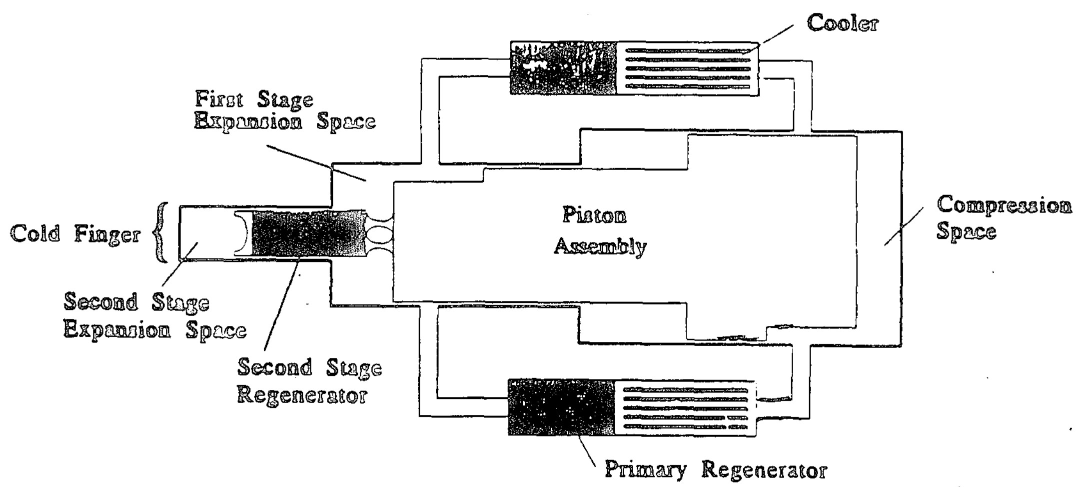



Figure 2.3. Two-stage Sibling Cycle refrigerator.

Because the second stage piston is connected to the first stage piston, its motion is the same. Because the first stage expansion space is continuously connected to the second stage expansion space, pressure in the second stage expansion space is the same as the pressure in the first stage expansion space except as altered by pressure drop in the second stage regenerator and freezer tube. Therefore, the pV diagram in the second stage expansion space must be essentially the same as that in the first stage expansion space unless pressure drop in the second stage regenerator and freezer tube is substantial.

2.4 THE FIRST STAGE

The first stage Sibling Cycle refrigerator hardware used in this project was originally designed for experiments with high-temperature, low-lift refrigeration. Design was predicated upon operating speeds from 10-20 Hz and mean pressures from 2 -10 MPA. Major dimensions and operating conditions are shown in Table 2.1.

Table 2.1. Major design parameters.

| | |
|---|---|
| Piston length (mm) | 257 |
| Bore (mm) | 42 |
| Stroke (mm) | 15 |
| Radial clearance (mm) | 0.0075 |
| Piston rod diameter (mm) | 12.7 |
| Pressure range (MPa) | 2-10 |
| Speed range (Hz) | 10-20 |

The piston is hollow at each end, with the fluid passing through a 4 mm round port in the cylinder wall through a 4 mm wide sinusoidal slot in the piston wall and thence into the expansion or compression space. Figure 2.4 is a photograph of the first stage piston.

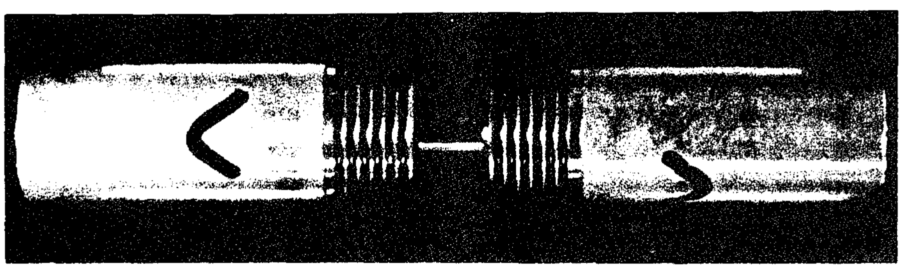


Figure 2.4. First stage piston.

2.5 THE DRIVE SYSTEM

The gearbox drive simultaneously rotates and reciprocates the piston. The gearbox is double-ended with 12.7-mm stainless steel piston rods protruding from both ends. One piston rod passes through seals into the operating space of the compressor/expander unit and is attached to the piston by a small-diameter flexible steel rod. The other piston rod carries a counterweight that balances the mass of the piston and piston rod and minimizes vibration. Exclusive of piston rods and external fittings, the drive box is 298 mm long.

The piston rods are mounted on double-acting thrust bearings attached to the connecting rods with wrist pins. The piston rods carry gears that are connected in train with the

crankshaft so that they rotate once for each four reciprocations, providing the necessary motion for alternate porting of the two sets of heat exchangers. The crankshaft is driven through a gear belt by a Leeson 370 W DC motor controllable from 0 to 29 Hz, initially geared 4 to 3. During the contract, the gearing was changed to 2 to 1, thereby limiting speed to 15 Hz but improving torque. Figure 2.5 is a photograph of the gearbox.

Initially, the drive box was designed to operate in a horizontal position, with the main journal bearings lubricated by a splash system and the other bearings lubricated by oil picked up from the bottom of the box and thrown around by the rotating gears. Later, the box was re-oriented in a vertical position with the counterweight on the bottom and the compressor/expander unit on top. That required installation of a pumped lubrication system using a GE 185 W motor driving a gear pump through a 5:1 reduction worm drive.

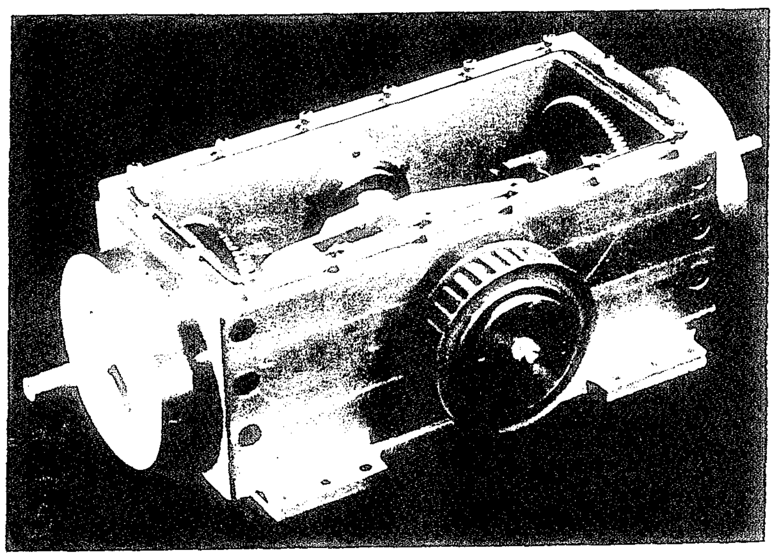


Figure 2.5. Double-acting gearbox drive.

## 2.6 CONFIGURATION OF THE COMPRESSOR/EXPANDER UNIT

The compressor/expander unit was initially housed within two bell-shaped castings held together by the cylinder heads and directly by bolts. The cylinder heads were threaded onto the cylinder at each end. Each bell-shaped housing contained a water-jacketed heat exchanger. The regenerators lay between the two housings. The connection between cylinder ports and heat exchanger ducts was through a Teflon ring of wedge-shaped cross section, held in place by wave springs. The housing was sealed with "O" rings, both externally and between the two separate sets of internal flow passages. The compressor/expander unit was 349 mm long.

The entire test rig, including gearbox drive, compressor/expander unit and second stage is shown in Figure 2.6.

The first stage piston and cylinder were both made of stainless steel for maximum dimensional stability. To provide an acceptable wear couple, the piston was coated with a baked Teflon coating. To minimize piston blow-by, the cylinder was honed and the piston ground to a nominal radial clearance of approximately 0.0075 mm. That clearance probably increased somewhat during the experiments as a result of wear.

The cylinder heads and housing castings of the original configuration were penetrated at 12 places by instrumentation ports. The dimensions of the ports were dictated by the use of Texmate MP-40A pressure transducers. The instrumentation ports were bored and tapped so that the transducers could be inserted into any port. The transducers fit flush with the inner walls of the housings, so that they did not alter dead volumes. The transducers were held in place by "O" rings backed by threaded brass glands. Unfortunately, this method of mounting the transducers appears to have created stresses in the transducers that altered their readings slightly.

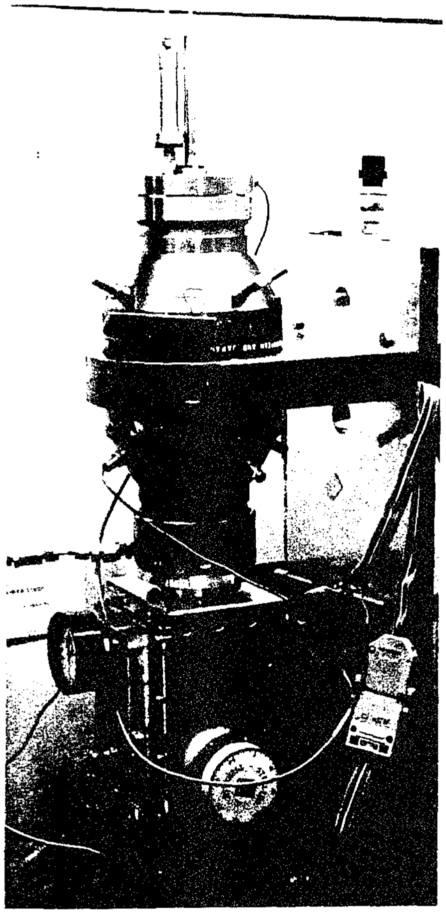


Figure 2.6. Complete test rig including first and second stages.

Brass plugs of similar dimensions were fabricated to fill the instrumentation ports not occupied by pressure transducers. To charge the compressor/expander with fluid, a brass plug was drilled and attached to a three-way valve. Several thermocouples ("E"-type) were potted in ceramic tubes which were then fitted into drilled brass plugs. The rest of the brass plugs were left blank. These arrangements permitted the compressor/expander to be instrumented at any of a dozen points with either a thermocouple or transducer. When the housings were later replaced by copper by-pass tubes, those opportunities for instrumentation were sacrificed.

The piston was threaded onto the piston rod and held in place by a lock nut tensioned by four set screws. During assembly, the piston was rotated onto the rod until it was a measured distance from the cylinder end with the crankshaft at top dead center. Slight rotational adjustment was then made so that a piston port was just tangent to a cylinder port (i.e., about to open) at top dead center. The piston was then locked in place.

The compression end cylinder head of the compressor/expander unit was bolted to a flange on a boss mounted on the end of the drive box. The piston rod passed through the center of the boss. The flange was slotted so that the bolt position could be adjusted through about 45 degrees of arc. That permitted the whole compressor/expander unit to be rotated relative to the drive box (and thus relative to the piston) to adjust port timing while the unit was fully pressurized, without disassembly.

Before thermodynamic testing began, the pressure containment had been filled with hydraulic fluid, refrigerated to about -5 C and pressurized to 20 MPa. The test included the heat exchangers. The highest pressure actually used in refrigeration experiments was less than 5 MPa.

## 2.7 GAS BEARINGS

The first stage was designed to operate with a combination of hydrostatic and hydrodynamic gas bearings. During the exchange stroke (as the piston moved from the compression end to the expansion end of the cylinder), fluid compressed in the isolated heat exchangers was intended to leak from opposed ports at both ends of the cylinder, energizing the hydrostatic bearing effect. The piston's rotation was intended to energize the hydrodynamic gas bearing.

The effectiveness of these two gas bearings is discussed in Subsection 6.1.3.

## 3.0 MODELLING THE TWO-STAGE EXPERIMENT

The first task performed under this contract was to model a two-stage Sibling Cycle refrigerator on computers preparatory to designing the second stage to be fitted to the existing first stage. That task was performed using two different codes.

Later, various versions of two-stage Sibling Cycle cryocoolers were modelled for possible use in Phase II. That effort was not constrained by any of the dimensions of the existing hardware. A promising two-stage design with stepped first stage piston was developed.

### 3.1 MODELLING WITH THE MS*2 CODE

The MS*2 Stirling Cycle code was created as an aid to development of the Sibling Cycle concept. It also models Stirling Cycle engines, refrigerators and cryocoolers, which permitted it to be validated against the published literature on a variety of machines with known performance. It has also been used to model a Stirling cryocooler developed by Hughes Aircraft for the United States Air Force using Hughes' proprietary data. The results show an extremely good fit between Hughes' performance data and the code prediction.

The MS*2 code is written in FORTRAN and runs on personal computers with 386 or faster microprocessors equipped with math coprocessors. It is a fully explicit code. It converges to an exact solution but the memory capacity and processor speed of personal computers limit the resolution that can be achieved on PCs. The code has been described in several publications (Refs. 6, 7, 10, 11, 12).

A unique feature of the code is its ability to model the leakage at cylinder ports of a Sibling Cycle machine based upon the number and dimensions of ports, piston/cylinder clearances, piston skirt lengths and type and temperature of working fluid.

The MS*2 code was used to model the first and second stages separately. It previously had been used to design the high-temperature refrigerator that served as the first stage in the experiments performed in this project. However, the code does not permit concurrent modelling of more than one expansion space. Two-stage modelling thus proceeded in a "piggy-back" mode, with the first and second stages being sized in such a way that the refrigeration generated in the first stage would "carry" the heat rejected in the second stage. In all, over 200 cases were modelled.

The second stage was modelled with the code as a conventional Sibling Cycle machine that experiences the same pV relationship in its expansion space as that generated by the (much larger) first stage. The requirements for compatibility between first and second stage models were thus:

- that both stages have the same piston stroke;

- that both stages share the same mean pressure and pressure ratio;

- that both stages operate at the same speed;

- that the heat rejection temperature of the second stage match the heat absorption temperature of the first stage; and

- that refrigeration in the first stage equal heat rejection in the second.

## 3.2 MODELLING WITH THE YARON CODE

Two-stage Sibling Cycle cryocoolers were also modelled with a code furnished by consultant Ran Yaron. More than 100 combinations of dimensions and operating conditions were tried.

The Yaron code is a "second order" code written in BASIC and run from the source code through the BASIC compiler. It permits all of the dimensions for both stages to be specified and then calculates ideal pV performance for both stages for specified operating conditions simultaneously. The result is then adjusted for pressure drop, regenerator ineffectiveness, static conduction and shuttle loss at each stage. The conduction loss in the second stage is treated as a gain in the first stage, since heat conducted from the first stage to the second stage enhances cooling in the first stage.

Optimization of the two-stage machine was predicated on determining the lowest temperature achievable in the second stage for specified first stage expansion space temperatures with heat rejected by the first stage at 300 K. For those conditions, the Yaron code predicted a second stage displacer/regenerator combination capable of achieving over 2 W of net cooling in the second stage at 70 K with maximum pressure under 4 MPa and speed of 20 Hz .

During experiments with the test rig, it became apparent that some of the assumptions about appropriate pressures and speeds were wrong. Optimum results appeared to occur with pressures below about 2 MPa and speeds below 12 Hz. Modelling had assumed higher pressures and speeds.

## 3.3 CONCLUSIONS FROM THE MODELLING EFFORT

The design of the second stage was based upon results from both computer models. The diameter of the second stage cylinder and the diameter and length of the regenerator housing were chosen on the basis of optimization studies using both the MS*2 and Yaron codes. Slight adjustments were made to permit use of stock sizes of seals and regenerator screens. The adjusted design was modelled and found to be satisfactory.

However, underlying all of the calculations was the assumption that the pressure ratio predicted for the first stage was realistic. As it turned out, the first stage pressure ratio was considerably below what had been expected.

When the second stage models were adjusted to reflect lower pressure ratios, predicted performance declined significantly. A table of results from selected second stage cases modelled with the MS*2 code is in Appendix A.

# 4.0 DESIGN AND CONSTRUCTION OF THE SECOND STAGE

The basic dimensions for the second stage were determined by the computer models, subject to various practical constraints. With the known constraints in mind, the second stage hardware was designed, fabricated and assembled in just 2 months. It proved to be reliable and adaptable.

## 4.1 SECOND STAGE DESIGN

The second stage was mounted on the cylinder head of the first stage. The second stage cylinder was coaxial with the main cylinder and the second stage piston was an articulated extension of the main piston. To connect the second stage piston to the first stage piston, a new plug system for the first stage piston was designed and fabricated. To facilitate assembly of the second stage, a new two-piece cylinder head was designed and fabricated. The butt end of the second stage cylinder was sealed to the cylinder head with an "O" ring.

The second stage regenerator housing was closed with a bolted flange and an "O" ring to permit easy assembly and modification. The second stage regenerator consisted initially of 1100 screens of 400 mesh stainless steel. Based upon the weight of screens, fill factor was estimated at about 0.28. Figure 4.1 illustrates the arrangement.

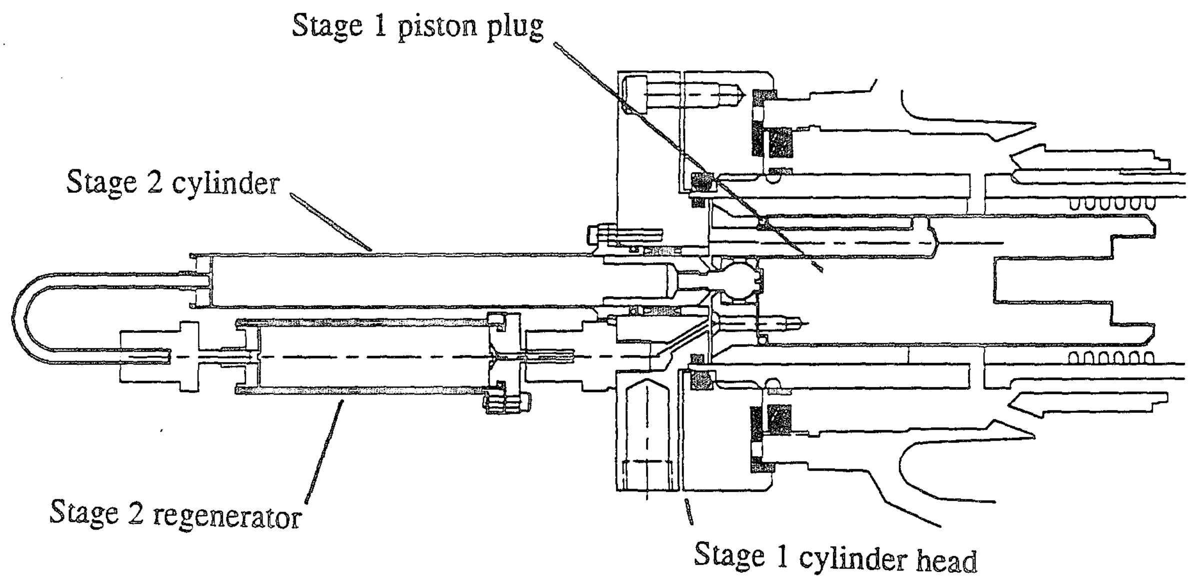



Figure 4.1. Second stage assembly.

The first stage was designed for a dead volume of 18.7 cm^3 in its expansion space. All of the dead volume of the second stage, including void volume in the second stage regenerator, had to be accounted for within that available dead volume. A minimum of about 7.3 cm^3 was required for ducts and clearances in the first stage itself, leaving about

11.4 cm^3 for the void volume of the second stage and all second stage ducts, manifolds and clearances.

The second stage regenerator was 60 mm long, 15.875 mm in diameter, with a fill factor of 0.28. That regenerator housing thus contained 8.55 cm^3 of dead volume. Piston diameter for the second stage was 12.7 mm, with clearance at top dead center of less than 1 mm, depending upon first stage piston position, which was adjustable. The dead volumes are described in Table 4.1

## 4.2 FABRICATION AND INSTALLATION OF THE SECOND STAGE

Integration of the second stage with the existing first stage Sibling Cycle cooler required several modifications of the first stage. The original first-stage piston design called for a displacement plug mounted on the expansion end cylinder head; the piston skirt moved in the annulus between the cylinder and the plug. That plug was redesigned to move with the piston, providing an anchorage for the driven end of the second stage piston.

Table 4.1. First and second stage dimensions and volumes.

| | |
|---|---|
| Swept in 1st Stage (mm^3) | 20782 |
| Unswept in 1st Stage (fraction) | 0.9 |
| Available dead volume | 18704 |

1st Stage Requirement

| Component/ Section | No. (mm) | L. (mm) | D. (mm) | Vol. (mm^3) |
|---|---|---|---|---|
| Duct (radial) | 2 | 3.3 | 4.5 | 105 |
| Duct (axial) | 2 | 58 | 7.13 | 4632 |
| Piston worms | | | | 1560 |
| Piston clearance | 1 | 0.7 | 42 | 970 |
| Total | | | | 7266 |

Second Stage Available

| | |
|---|---|
| (Available dead volume less total 1st stage requirement) | 11437 |

Second stage components

| Component/ Section | No. (mm) | L. (mm) | D. (mm) | Vol. (mm^3) |
|---|---|---|---|---|
| Regen;(.72 void) | 1 | 60 | 15.875 | 8551 |
| Regen. duct | 1 | 42 | 2 | 132 |
| Regen. manifolds | 2 | 0.8 | 15.875 | 317 |
| Fitting plenum | 1 | 5 | 7.5 | 221 |
| Slant hole | 1 | 11 | 2.8 | 68 |
| Piston clearance | 1 | 3 | 12.7 | 380 |
| Piston annulus | 1 | 60 | 12.7 | 119 |
| Freezer | 1 | 100 | 1.6 | 201 |
| Cyl. head bore | 1 | 2 | 12.7 | 253 |
| Fill port | 1 | 20 | 2.8 | 123 |
| Total | | | | 10365 |

The necessary second stage components were designed and fabricated primarily from stainless steel. The principal components initially constructed for the second stage were the second stage piston, cylinder, regenerator housing, and new first stage cylinder heads (two pieces).

To accommodate the second stage, the new two-piece expansion-end cylinder head was machined. To seal the end of the first stage cylinder, an internal "O" ring seal was provided. (Originally, sealing had been accomplished with an external "O" ring.) The outer part of the two-piece cylinder head was machined to accept the second stage cylinder, with second stage piston seal, and the fitting for the connection to the regenerator.

Access was also provided for a plug fitting that could be, interchangeably, a pressure transducer, a thermocouple or a supply port for helium.

The outer end of the second stage cylinder was connected to the outer end of the second stage regenerator with bent stainless steel tubing, which served as the freezer, and by commercial tube fittings. The inner end of the regenerator housing was connected to the new cylinder head with a commercial tube fitting. Provision was made to enclose the second stage in a vaccum dewar if warranted by test results.

The second stage piston was fabricated from linen-filled phenolic resin press fitted into a stainless steel shank that couples it to the first stage piston with a ball joint.

The stage cylinder walls are 0.257 mm thick. The tube fitting that connects the warmer end of the second stage regenerator to the first stage cylinder head is 3.14 mm in diameter, with a 1.6 mm ID.

Figure 4.2 is a photograph of the second stage cylinder, freezer tube and regenerator housing attached to the new first stage cylinder head.

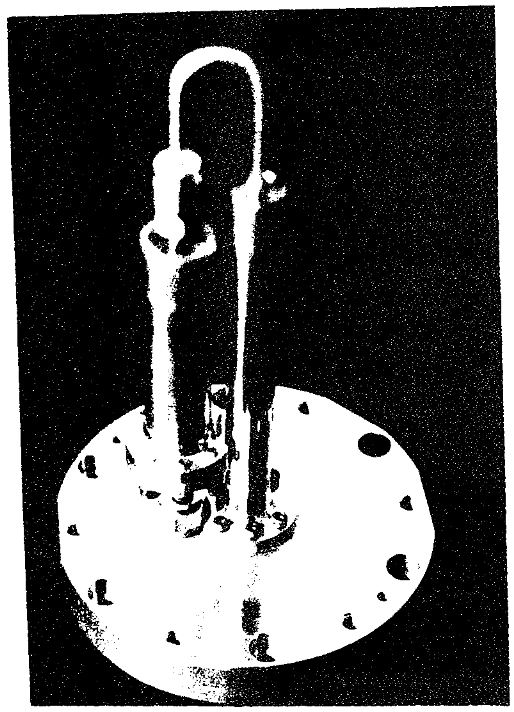


Figure 4.2. Photograph of the second stage.

## 5.0 EXPERIMENTS WITH TWO-STAGE SIBLING CYCLE HARDWARE

The experimental program combined hardware experiments with computer modelling and theoretical analysis on an iterative basis. Over the course of 4 months, 15 hardware modifications were conceived, executed and evaluated. During that effort, numerical data were taken for over 130 test runs at a variety of pressures and operating speeds, primarily with He as the working fluid (see Appendix B). The primary objectives of the testing program were to develop and evaluate pV data for the first stage and to improve the cooling rate in the second stage.

### 5.1 TEST PROCEDURES

Data acquisition was through a Keithley Metrabyte DAS16G A/D data acquisition board installed in a PC with an Keithley Metrabyte external EXP16 board and EASYEST LX data analysis software from the same supplier. Texmate MP-40A pressure transducers were inserted in various locations in the machine, including both sides of the regenerator, both heat exchanger assemblies and both cylinder heads at various times.

Type "E" thermocouples were likewise placed in various locations. As many as eight thermocouples could be monitored simultaneously, but most experiments were conducted with fewer. Two pressure transducers were available and could be used simultaneously in parallel with thermocouples. The dimensions of pass-through fittings for the thermocouples were the same as the pressure transducer dimensions, allowing great flexibility in locating sensors at various places in the machine.

The first stage was equipped with a Talley LD600-15 linear transducer. The transducer was mounted to follow the position of the counterweight drive shaft of the gearbox drive. Since the counterweight moves in exact opposition to the piston, that arrangement made it possible to generate pV diagrams rapidly and accurately by integrating data acquired from the pressure transducers with the position (i.e., volume) output of the linear transducer.

Initial thermodynamic testing was done with the refrigerator axis in the horizontal position. Re-orientation of the machine in a vertical position proved beneficial. The mass of the piston exceeds 1.0 kg. Although both hydrodynamic and hydrostatic gas bearing effects had been posited, it was not clear that either effect was being achieved in the horizontal orientation. Subsequent analysis suggests that the hydrodynamic bearing effect would have been inadequate to support the piston's weight and that the intermittent hydrostatic bearing effect would not have replaced it.

For the initial experiments in the horizontal position, the power input to the drive motor was monitored with an induction coil clamped to the power cord. Fluctuating readings were obtained, with the period of fluctuations ranging from seconds to minutes. Some variation in mechanical friction was suspected, and on several occasions, power was shut off when the power input readings began to escalate. When the machine was remounted in the

vertical position, power fluctuations diminished and monitoring of power input was discontinued.

The refrigerator was charged with either dry N2 or He. The N2 was used after each mechanical change until the refrigerator appeared to be operating satisfactorily. The N2 was then replaced with He, with several cycles of purging, before experiments were conducted. As discussed below, however, the purging may not have been adequate.

Numerical results of the test program are summarized in the tables in Appendix B. The discussion below supplements that material with details of the hardware modifications made during the testing program and discussion of the results obtained.

## 5.2 MODIFICATIONS OF HARDWARE

Testing began with motoring tests on June 29, 1993. Timing was adjusted and an approximately correct pV diagram for the first stage was obtained on July 5. However, no significant cooling was observed in the second stage.

Various combinations of port timing, pressure and operating speed were tried on July 9 and 12 . Some slight cooling tendency (1.4 K in 10-14 minutes) was observed at low speeds and pressures, (e.g., 5 Hz, 1.5 MPa) but under other conditions, the second stage freezer merely held its temperature, or actually became warmer.

At the end of the July 12 session, the second stage regenerator was removed and replaced with a coil of copper tubing about 1 m long and 1.6 mm ID. That seemed to work better than the screen regenerator although axial conduction was undoubtedly high and the regenerative effect low.

Typical pressure ratios with the machine as originally configured were about 1.11. Computer modelling had predicted pressure ratios of 1.2. Because the measured pressure ratio was low relative to predictions based on computer models, efforts were made to increase the pressure ratio. The first step in that direction was to build a new first stage regenerator using a spare regenerator blank, spare screens and #9 lead shot. The shot were ultrasonically cleaned, then baked briefly to dry them. The screens were used only to retain the shot in position in the regenerator. Diameters of the shot ranged from about 1.8 to 2.2 mm. Based upon the weight of shot used (1,150 g), the fill factor was calculated as 0.54 versus 0.28 for the original regenerator with 250 mesh screens. That increased the pressure ratio to 1.16 still well below computer predictions.

Further tests on July 15 continued with the bare copper tube as second stage regenerator and the high-density lead shot regenerator in the first stage. Cooling rates of 1 K in 3 minutes were obtained. That was an improvement, but still unsatisfactory. When the original, screen-packed second stage regenerator was replaced, performance declined.

On July 19, the pressure ratio was further increased to 1.16 to 1.18 (depending upon port timing) by filling a duct with bearing balls. The increased pressure ratio did not appear to improve performance with the original screen-packed second stage regenerator in place. No significant progress was made during runs on July 24.

On July 27, the copper tube second stage regenerator was re-installed. With the other modifications in place, a pressure ratio of 1.2 was regularly obtained. Best cooling performance was a drop of 1.4 K in 235 seconds, or 0.36 K per minute. That was only a slight improvement on the performance obtained with a lower pressure ratio. That result was obtained at 11 Hz with a maximum cycle pressure of 1.8 MPa.

On July 28, the original second stage regenerator was re-installed and a series of runs made at the conditions that had been found favorable with the copper-tube substitute regenerator. No cooling was observed. The second stage freezer temperature rose from about 0.15 K to 0.25 K per minute, with the largest temperature rise associated with low speed operation in the range of 5 Hz. During all of those runs, the first stage pV diagram showed substantial potential refrigeration.

Later on July 28, the original second stage regenerator was altered by removing 10 mm of screens and replacing them with #9 lead shot 1.8 - 2.2 mm in diameter. The effect was to alter the overall fill factor slightly and to reduce pressure drop in the second stage regenerator. A slight reduction in pressure ratio was observed. Runs on July 28 and July 30 seemed to show some reduction in the rate at which the second stage freezer was warming, to about 0.16 K per minute.

The pV diagrams for the first stage were recorded on July 30. Figure 5.1 is a plot of pressure versus piston position in the expansion space of the first stage. The plot reads clockwise in the large loop. Reciprocating speed was 9 Hz, maximum pressure about 1.72 MPa. The pressure ratio of 1.19 was considerably higher than the 1.09 to 1.11 ratios seen in earlier runs. Timing was good; the sharp corner at maximum pressure and minimum volume indicated that the fluid in the expansion space was pre-compressing to the same pressure as the isolated heat exchangers and that the expansion end ports were opening cleanly at top dead center.

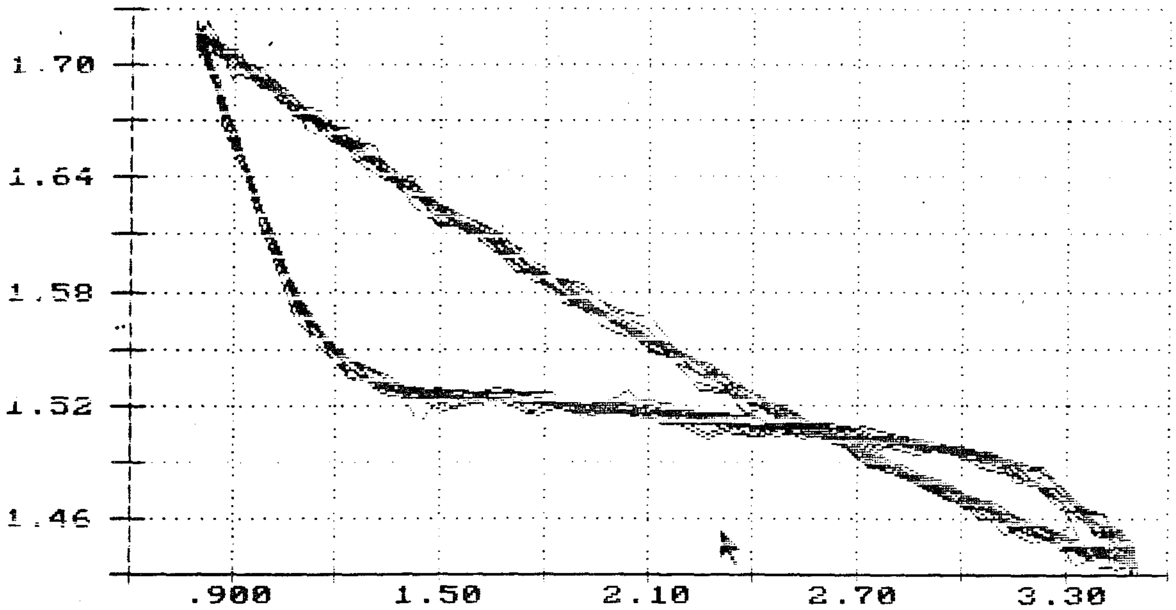



Figure 5.1. The pV diagram, first stage expansion space, 7/30/93, run #5.

Good timing at the expansion end of the machine was achieved without spoiling the timing at the compression end, as shown by Figure 5.2.

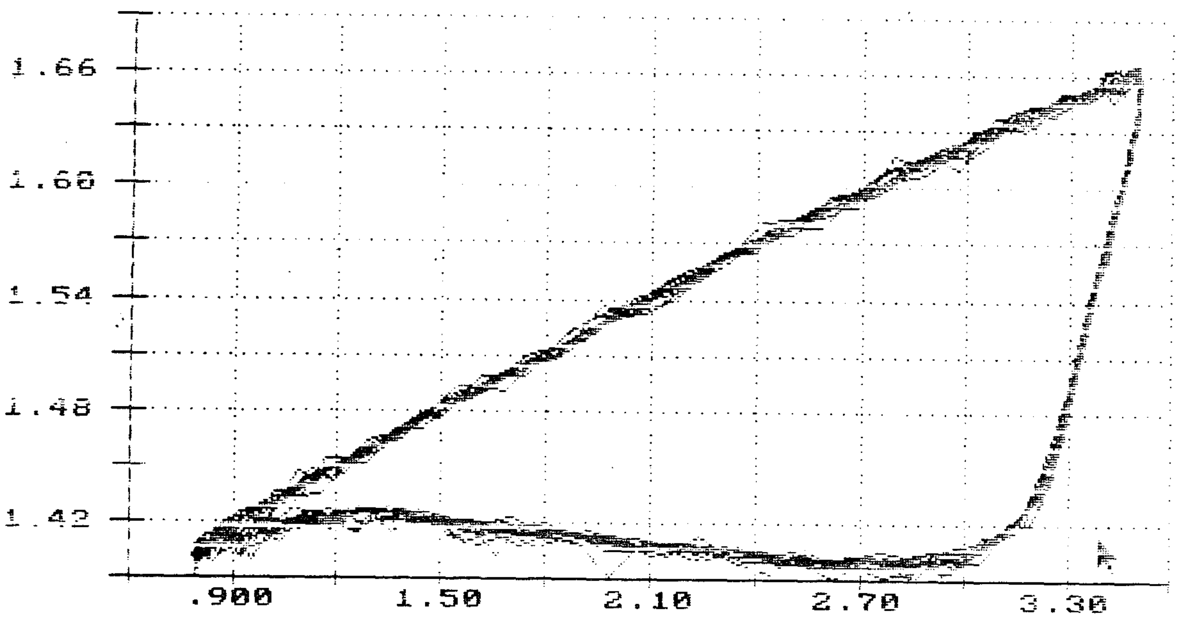



Figure 5.2. Pressure/position diagram, compression end, 7/30/93, run #5.

Figure 5.2 shows a pressure-position trace for the compression end of the first stage. It is the mirror image of a pV diagram; the pressure transducer reads position for the whole piston, but the volume changes at the ends of the piston are opposite. The plot reads clockwise.

At minimum compression space volume (to the right side of Fig. 5.2) compression ends with a sharp drop in pressure as the piston moves away from the bottom dead center position with all ports still closed. At the bottom of the descent, compression end ports open to low pressure. During the middle of the transfer stroke (across the bottom of the trace), pressure rises slightly as fluid leaks in from the isolated heat exchangers and as the fluid moves from a cooler space to a warmer space. As the piston approaches top dead center (to the left side of Fig. 5.2) ports close at the expansion end, terminating the inflow of fluid to the compression space. However, the piston continues to move, increasing the volume in the compression space and dropping pressure close to the low point of the cycle. Finally, the compression stroke shows a steady rise in pressure from the lower left corner to the upper right corner of Figure 5.2.

Despite the promising pressure/position plots, the second stage produced no cooling.

Throughout the tests to this point, the measured pV plots had shown substantial potential cooling in the first stage expansion space. As pressure ratio increased, the large loop area of the pV diagram that represents cooling grew. However, the small loop representing losses grew even more rapidly, probably reflecting less satisfactory regeneration in the first stage as lead balls were substituted for a much finer screen. The increased pressure swing may also have enhanced hysteresis losses.

By the end of July, it appeared that efforts to increase first stage pressure ratio might have been less effective than efforts to reduce the pressure drop in the second stage regenerator.

During August, data were taken for 88 experimental runs using 5 different second stage regenerator configurations in 9 test sessions. The perplexing feature of the month's effort was that promising pV diagrams were consistently obtained for the first stage, but cooling remained feeble in the second stage.

Another curious phenomenon was observed. At startup, temperature in the freezer tube rose initially, usually by about 0.5 K, before settling down and beginning slowly to decline. Then, after shutdown, the temperature continued to fall for several tenths of a degree (K) over a period of several minutes. These phenomena may be explained by reference to helium leakage.

Because the test rig had many seals and fittings, it was not possible to completely prevent leakage. To allow for constant-pressure test runs, a check valve was inserted in the line between the helium bottle and the compressor/expander unit. When the machine was at rest, piston blow-by and port leakage soon brought the interior of the entire pressure vessel to a common pressure set by the regulator. However, when the machine was started, the

pressure began to fluctuate above and below mean pressure and at each low point in the pressure cycle, additional He entered through the check valve. The effect was to make the regulator pressure the minimum pressure during operation instead of mean pressure, as it was with the machine at rest.

The unusual temperature behavior is thus accounted for by the adiabatic increase in temperature as additional fluid entered the system at startup and the corresponding adiabatic decrease in temperature after the machine was shut down and mean pressure leaked down to the regulated level.

That phenomenon, however, does not explain the relatively small refrigeration generated in the second stage. One possibility under consideration was that the second stage regenerator was creating too much pressure drop through 60 mm of 400 mesh screens. During August, the second stage regenerator was rebuilt several times with coarser matrix materials (spheres) replacing progressively larger portions of the 400 mesh screens. The improvement in performance was slight. In its most open configuration, the second stage regenerator demonstrated cooling performance only slightly better than was obtained when a 500 mm length of 1.6 mm OD copper tube was spliced into the regenerator's space in place of the stacked screen regenerator.

Another possible explanation was that the pressure ratio generated by the first stage still was not adequate. During the testing performed under this contract, several maneuvers were attempted to increase the pressure ratio. One was to decrease the dead volume in the duct between the first stage cooler and the cylinder wall port by packing it with stainless steel ball bearing balls. That had a slight effect on pressure ratio and had little or no positive effect on second stage performance. The balls were removed late in August.

An alternate approach was to drastically reduce the volume of the first stage regenerator. That was accomplished by fabricating a new regenerator blank. The re machined blank was installed (without screens) at the end of August and was in place when the best cooling rates of the month were obtained. Pressure ratios of 1.21 were routinely observed. Experiments seemed to show some improvements in cooling.

During September, promising pV diagrams were consistently obtained for the first stage, but cooling was feeble in the second stage. It began to appear that cooling in the end of the second stage cylinder and the cold end of the second stage regenerator was being lost in the connecting tube (75 mm long, 1.6 mm ID). The possibility of adverse Joule-Thomson effects was considered. At the test temperatures, the Joule-Thomson coefficient of He is negative, but the coefficient of N2 is positive. However, other properties of nitrogen (especially gamma) made it impractical to confirm that theory by substituting N2 for He. With N2, there was no cooling at all.

By mid-September, ideas for minor alterations appeared to have been exhausted. Radical changes in the first stage, made as part of MS*2's internal R&D, were therefore combined with the second stage cylinder/regenerator assembly. The first stage changes were

fundamental: the bell-shaped housings, heat exchangers and regenerators were removed. Steel collars were clamped to the bare cylinder around the ports at each end. The ports at expansion and compression end, respectively, were connected with 4.6 mm (ID) copper tubing using Swagelok tube fittings. The effect of this change was to significantly increase the pressure ratio of the first stage by reducing the unswept volume in the space between the expansion and compression end ports. It also drastically altered the heat transfer regime. The copper tubes functioned primarily as heat rejecting heat exchangers. No formal regenerator remained. No formal heat absorbing heat exchanger remained.

After initial testing without the second stage, the second stage was again put in place on the modified first stage. Pressure ratios as high as 1.4 were predicted by the MS*2 code. The highest ratio repeatably achieved was 1.3. Test results were perplexing. On October 26, cooling rates of 0.5 and 0.48 K per minute were achieved in 5 minute runs at 11.5 Hz and 0.95 MPa. On the following day, runs at slightly lower pressure and speeds both higher and lower than 11.5 Hz produced essentially no cooling in the second stage. The final runs were completed that day.

## 6.0 ANALYSIS OF DATA OBTAINED

The first and most important question to be addressed is why the experiment did not produce higher rates of cooling. Although the codes predicted relatively modest cooling at the speeds and pressures ultimately selected as most promising, the cooling actually measured fell below those levels.

For simplicity in the testing procedure, cooling was measured in terms of temperature changes in various parts of the machine. No heat was added or removed except by natural convection and conduction through the drive box, gas supply tubing and instrumentation wires. The runs were normally of short duration and the temperature excursions above and below ambient were relatively small. In the circumstances, conduction and convection were disregarded.

Figure 6.1 illustrates temperature changes in different parts of the first and second stages over a series of consecutive runs made on September 9, 1993. Over 25 minutes of operation, temperature in the compression end duct in the first stage rose from 28.2 to 34 C while temperature at the top of the second stage cylinder dropped from about 23.8 to 20.7 C. Temperatures of the other parts of the second stage also trended down, but less sharply. The temperature of the first stage cylinder head ended the runs at essentially the same temperature as it had started at. The starting temperatures were different because the

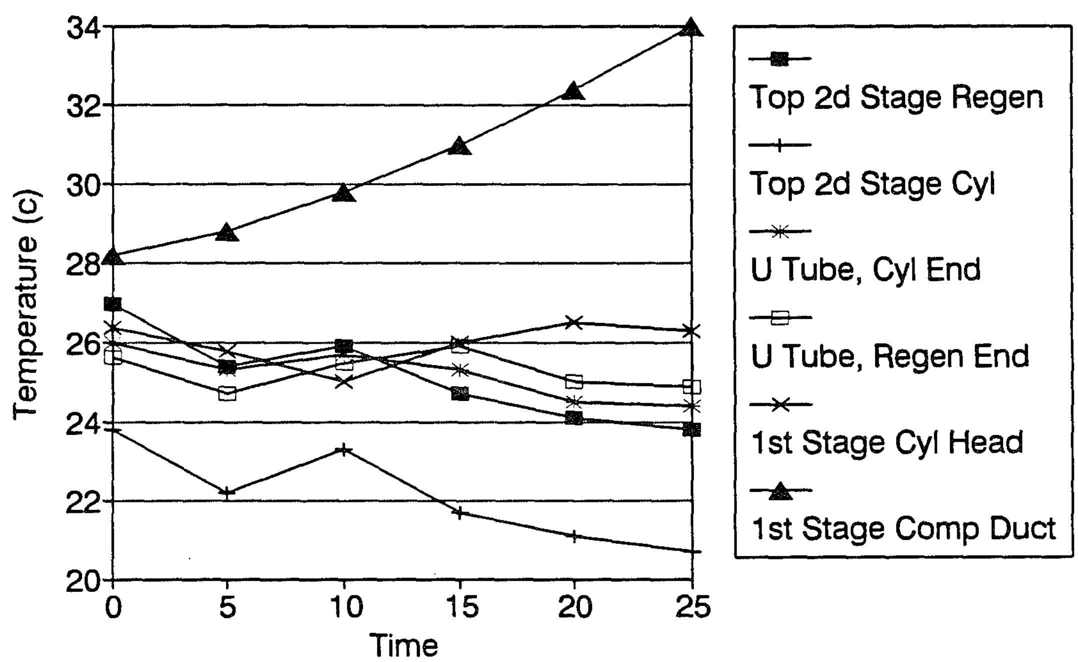



Figure 6.1. Temperature changes in different parts of the test apparatus.

machine had already been run several times when these data were accumulated. The final temperature difference between the compression end of the first stage and the expansion end of the second stage was slightly more than 13 K.

To determine the amount of heat being transferred at various locations in the machine, the thermal mass of the components of the first and second stages was calculated in terms of the number of Joules of heat required to be added or removed to change the temperature of the component by 1.0 K.

The thermal mass of the expansion end of the first stage as originally configured was about 6,100 J/K (Table 6.1). As a whole, it thus cooled relatively slowly; with 50 W of cooling in the first stage, the temperature would have dropped about 0.5 K per minute.

Table 6.1. Thermal mass of the first stage compressor/expander

| Piece Part | Material | No. of Parts | Spec. Heat (c/g/K) | Mass (g) (each) | Thermal Mass (J/K) Warm end | Cold end |
|---|---|---|---|---|---|---|
| | | | | | | |
| Cast housing | cast iron | 1 | 0.119 | 4637 | 2307 | 2307 |
| Heat exchanger | aluminum | 1 | 0.215 | 373 | 335 | 335 |
| Cylinder | stainless | 0.5 | 0.12 | 1622 | 407 | 407 |
| Piston | stainless | 0.5 | 0.12 | 1000 | 251 | 251 |
| Regen housing | Delrin/Kevlar | 0.5 | 0.4 | 363 | 303 | 303 |
| Regen matrix | stainless | 0.5 | 0.12 | 427 | 107 | 107 |
| Connector wedge | Teflon | 1 | 0.25 | 30 | 31 | 31 |
| Pusher ring | Delrin | 1 | 0.35 | 146 | 214 | 214 |
| Retainer nut | Delrin | 1 | 0.35 | 22 | 32 | 32 |
| Large wave spring | Steel | 1 | 0.118 | 20 | 10 | 10 |
| Small wave spring | Steel | 1 | 0.118 | 15 | 7 | 7 |
| Compression head | ledloy | 1 | 0.1 | 2622 | 1096 | N/A |
| Expansion head | ledloy | 1 | 0.1 | 2376 | N/A | 993 |
| Compr. cyl. plug | Delrin | 1 | 0.35 | 84 | 123 | N/A |
| Expansion cyl. plug | Delrin | 1 | 0.35 | 89 | N/A | 130 |
| Compr. piston plug | Delrin | 1 | 0.35 | 54 | 79 | N/A |
| Exp. piston plug | Delrin | 1 | 0.35 | 55 | N/A | 80 |
| Piston rod | steel | 0.5 | 0.118 | 57 | 14 | 14 |
| Piston rod nut | steel | 0.5 | 0.118 | 25 | 6 | 6 |
| Gearbox flange | aluminum | 1 | 0.226 | 347 | 328 | N/A |
| Gas valve | brass | 1 | 0.092 | 399 | N/A | 153 |
| Plugs w/gland | brass | 4 | 0.092 | 40 | 62 | N/A |
| Plugs w/gland | brass | 6 | 0.092 | 40 | N/A | 92 |
| Transducers w/gland | stainless | 1 | 0.12 | 25 | 13 | 13 |
| Housing bolts | steel | 2 | 0.118 | 20 | 20 | 20 |
| Gearbox bolts | steel | 4 | 0.118 | 40 | 79 | N/A |
| Potting | Woods met. | 4 | 0.04 | 880 | N/A | 589 |
| TOTAL | | | | 15808 | 5823 | 6095 |

By contrast, the thermal mass of the entire second stage ranged from about 110 to 130 J/K (Table 6.2). Thus, just 1 W of cooling would have reduced the temperature of the entire second stage by almost 0.5 K per minute. In fact, that rate of cooling was observed only once at any single point in the second stage assembly. (A summary of all the runs that produced meaningful numeric data is in Appendix B.)

Table 6.2. Thermal mass of second stage components.

| Piece Part | Material | Spec. Heat (c/g/K) | Mass (g) (each) | Thermal Mass (J/K) |
|---|---|---|---|---|
| | | | | |
| Cyl+tube+ftg | stainless | 0.12 | 44 | 22 |
| Piston | | | | |
| tip | linen phenolic | 0.35 | 18 | 26 |
| end | stainless | 0.12 | 23 | 12 |
| Reg hsg + 300 scr | stainless | 0.12 | 89 | 45 |
| TOTAL w/o matrix | | | 174 | 105 |
| Regen matrices: | | | | |
| Screens | stainless | 0.12 | 26.6 | 13 |
| Copper tube | copper | 0.092 | 20 | 8 |
| Screens/balls | steel | 0.12 | 52 | 26 |
| Screens/G10 | mixed | 0.25 | 21 | 22 |
| TOTAL with: | | | | |
| Screens | | | | 118 |
| Copper tube | | | | 112 |
| Screens/balls | | | | 131 |
| Screens/G10 | | | | 127 |

In general, the temperatures in various parts of the second stage varied considerably, and at times one part would be warming up while another was cooling down. Thus it would be inappropriate to treat a temperature change at one point in the second stage as representative of temperature change in the whole second stage assembly. As a whole, the second stage assembly never approached a cooldown rate of 0.5 K per minute.

Computer analysis of second stage performance suggests that cooling rates should have been greater than actually recorded. For example, the MS*2 code projected about 0.44 W of cooling at ambient temperature at 10 Hz with an average charge pressure of 1.5 MPa of

pressure and pressure ratio of 1.1. That would have produced cooling of about 0.2 K per minute throughout the second stage. Although that cooling rate was attained at times in localized parts of the second stage, it is clear that other parts cooled more slowly, if at all.

The Yaron code generated somewhat more optimistic predictions of second stage performance than did the MS*2 code; the actual results were thus further from matching the predictions of the Yaron code.

### 6.1 POSSIBLE EXPLANATIONS FOR OBSERVED PERFORMANCE

The disparity between computer predictions and observed performance can be explained either of two ways: (1) the computer programs are overoptimistic or (2) the machine was not, in fact, as described to the programs. Both possibilities are considered.

#### 6.1.1 Validity of the Codes

Refrigerators based on the Stirling Cycle principle can be modelled on computers by a variety of methods. Accuracy of the results depends upon both a valid computational scheme and an accurate description of the machine being modelled.

Both the MS*2 and Yaron codes are mature. They have been used to model many Stirling Cycle coolers. They generally provide quite accurate predictions for well-described Stirling cooler designs (Refs. 6, 7, 10, 11, 12). They can be therefore considered valid.

#### 6.1.2 Accuracy of the Description

The two-stage Sibling Cycle refrigerator investigated under this contract was described to both codes as it was believed to be. Nevertheless, it is possible that one or more elements of the description was incorrect in a way that would tend to reconcile the observed results with the computer predictions.

Possible inaccuracies include the description of the piston/cylinder clearance, the composition of the fluid inside the machine and the true pV relationship in the second stage. All of these possibilities are discussed below.

#### 6.1.3 Piston/Cylinder Clearance

The first stage Sibling compressor/expander is especially sensitive to leakage in the clearance between piston and cylinder. Excessive clearance generates "blow-by" from one end of the piston to the other. But that was not the most serious problem. Because the piston and cylinder are both ported, leakage through ports was an even more significant source of loss. The MS*2 code models port leakage as a function of piston/cylinder radial clearance. If the radial clearance described to the code is wrong, the results of modelling will also be wrong. There are several reasons to believe that the modelling assumptions were optimistic and perhaps very optimistic.

The nominal radial clearance upon which most of the original calculations were based was 0.0075 mm. That is half the diametral clearance measured when the cylinder was honed and the piston ground. Any increase in that clearance has serious consequences for the performance of the system. There are several reasons to believe that the actual clearance in the machine was significantly greater than 0.0075 mm. If so, many of the observed phenomena become much clearer. Reasons for believing that the clearances are larger than 0.0075 mm include the following:

6.1.3.1 Expert Opinion

Consultation with, Ran Yaron, informal consultation with Dr. Peter Kerney of CTI Cryogenics, Mansfield, Massachusetts and Reuven Unger of Sunpower, Inc., Athens, Ohio, has evoked the unanimous view that excess leakage is a likely explanation for the unexpectedly low pressure ratios and poor refrigeration capacity.

Calculations prepared by consultant R. Warren Breckenridge of Arthur D. Little, Inc., (ADL), Cambridge, Massachusetts, show that any significant hydrodynamic gas bearing effect arising out of the rotation of the first stage piston would depend upon significant eccentricity of the piston relative to the cylinder; the hydrodynamic gas bearing effect is incapable of completely centering the piston under any circumstances. Thus, the radial clearance was necessarily different at different points on the circumference of the piston, and larger than the nominal 0.0075 mm at some points.

Figure 6.2 illustrates the relationship between eccentricity and bearing load. It shows that at typical operating conditions of 1.5 MPa pressure, 12 Hz reciprocating speed (3 Hz rotational speed) and existing dimensions, the gas bearing would support lateral loads on the piston of less than 1.0 N at an eccentricity of 0.4. Under those conditions, the piston would be 0.0045 mm from the cylinder on one side and 0.0105 from the cylinder on the other side. Although the piston was mounted to reciprocate vertically, it is by no means certain that lateral forces on the piston induced by pneumatic loads were less than 1.0 N; the eccentricity could have been significantly greater than 0.4.

If the piston is off center, the increase in leakage through the port with larger clearance is much greater than the reduction in leakage through the port with reduced clearance; leakage is proportional to the cube of the clearance gap.

Consultant Yaron has expressed doubt, based upon experience, that the original radial clearance was accurately measured in the first place. The outer diameter of the piston and the inner diameter of the cylinder were carefully measured by a reputable grinding shop with analog gauges (micrometers and borescopes). Straightness of the piston was checked on a granite slab. However, more accurate instruments might have detected out-of-round conditions of either piston or cylinder as well as deviations from straightness. Yaron believes that even if the piston and cylinder were originally round, straight, and the exact sizes measured, installation has undoubtedly subjected them to some stresses that may have

altered their dimensions significantly in relation to the nominal 0.0075 mm radial clearance. Pressurizing the cylinder also causes it to expand slightly, as well as creating forces that tend to distort its shape, further increasing clearances, at least at some points.

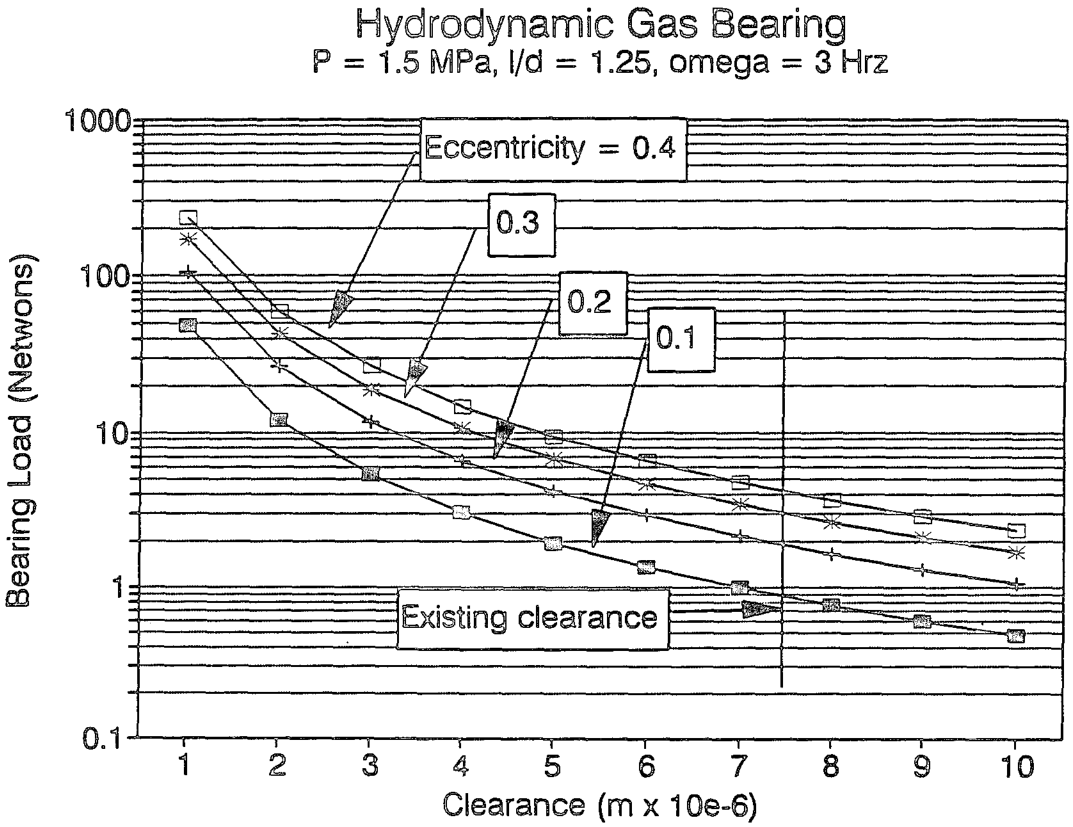



Figure 6.2. Hydrodynamic gas bearing support for the piston.

6.1.3.2 <u>Limitations of the Hydrostatic Gas Bearing</u>

At times during the cycle, high pressure fluid on the outboard sides of opposed ports may create a hydrostatic gas bearing that tends to center the piston. It is also possible that that pressure creates a "hydraulic lock" in which the piston is drawn toward a port from which high pressure fluid is leaking. The conditions which give rise to the hydrostatic bearing (or hydraulic lock, as the case may be) fluctuate over the cycle.

During the middle portion of the transfer stroke, the opposing ports on the "A" side (and, alternately, the "B" side) contain high pressure while both ends of the cylinder are at lower pressure. If the hydrostatic gas bearing works at all, it works during this portion of the cycle.

However, during the expansion/compression stroke, the picture is much more complicated. At the beginning of that stroke, pressure starts out high in the side ("A" or "B" side) that is connected to expansion space. The closed ports at the expansion end are open at the compression end and are thus connected to a low pressure volume. There is an apparent tendency for the piston to be pushed against one or the other of the closed ports on the expansion end, thereby increasing the gap on the other side of the piston and permitting some leakage from the high-pressure expansion end to the low pressure compression end through ports.

At the end of the expansion/compression stroke, pressure in the compression end of the cylinder is high while pressure in the expansion end is low. The closed ports at the compression end connect with the low-pressure expansion space and there is an apparent tendency for the piston to be pushed against one of those closed ports at the compression end, decreasing clearance at one port but increasing it at the other. Again, fluid would tend to leak through the low pressure port with the larger clearance.

The same tendency to instability of piston position continues after all ports close at the compression end at bottom dead center. While high pressure remains momentarily trapped in the cylinder, opposing closed ports (soon to be opened) are at lower pressure than the cylinder, tending to perpetuate the instability that developed at the end of the expansion/compression stroke.

By the same token, a similar instability may exist at the expansion end of the cylinder at the end of the transfer stroke, when high pressure is building up in the expansion space preparatory to port opening at top dead center.

It is apparent that, during portions of the cycle, any hydrostatic bearing effect created by high pressure fluid emerging from cylinder ports into the gap between cylinder and piston momentarily disappears. During these intervals, only the relatively feeble hydrodynamic gas bearing effect remains.

#### 6.1.3.3 Wear on the Piston

Inspection of the piston after several hours of operation showed evidence of burnishing on its Teflon-coated surface. Inspection of the inside of the cylinder shows characteristics witness marks of piston contact, tracing the piston's undulating path as it rotated and reciprocated.

There is no doubt that piston and cylinder made contact on many occasions - perhaps even continuously. If the piston was lying against a port on one side of the cylinder, it was at least 0.015 mm from the port on the other side, not 0.0075 mm from it.

Wear on the piston is not merely evidence of eccentricity. It also indicates that the diameter of the piston was shrinking as the Teflon coating wore off. That tended to increase the gap between piston and cylinder port, further enhancing leakage.

#### 6.1.3.4 Pressure Traces Taken During Operation

During experimental work with the original first stage configuration, pressure plots occasionally produced "twin" traces in which maximum pressure in the "A" side heat exchangers was persistently different from that in the "B" side. Figure 6.3 is an example. This phenomenon was rare, unusual and unexplained. Whether it was a further clue to piston position, and thus to piston/cylinder clearance, is unclear.

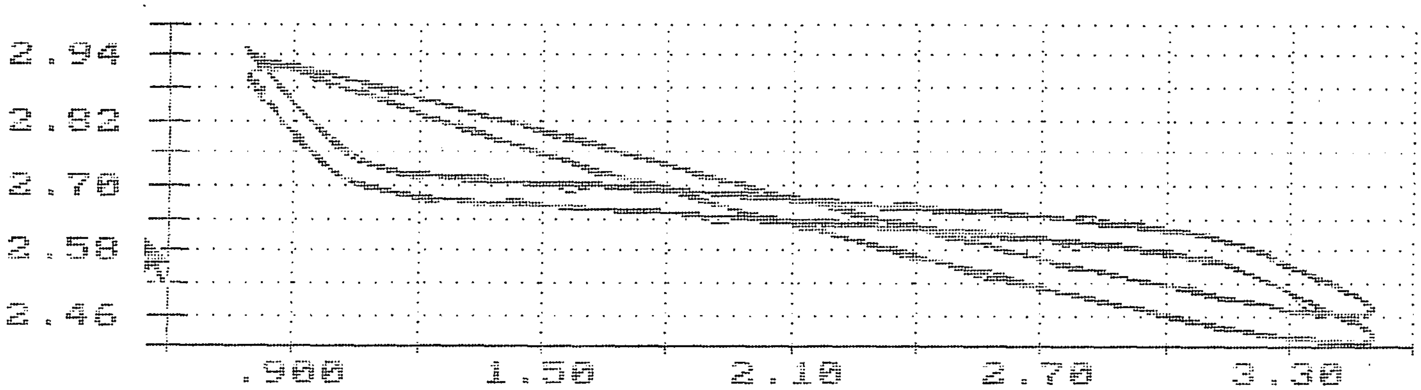



Figure 6.3. Oscillating expansion space pV diagram.

#### 6.1.3.5 Analysis with the MS*2 Code

The impact of excessive piston/cylinder clearance on MS*2 code results is dramatic. Figure 6.4 shows the predicted relationship between port clearance and first stage refrigeration for operation at 2 MPa, 13.33 Hz, heat absorption temperature of 290 K and heat rejection temperature of 300 K. Table 6.3 shows the relationship between clearance and other parameters (timing, pressure ratio) under the same conditions.

Table 6.3. Impact of piston/cylinder clearance on performance parameters.

| Clearance (mm) | Pressure ratio | Timing (expansion) | Timing (compression) | Refrigeration (W) |
|---|---|---|---|---|
| 0.007 | 1.36 | 40 | 47.25 | 29 |
| 0.015 | 1.38 | 35 | 43 | 5 |
| 0.02 | 1.35 | 27.5 | 34.25 | -19 |

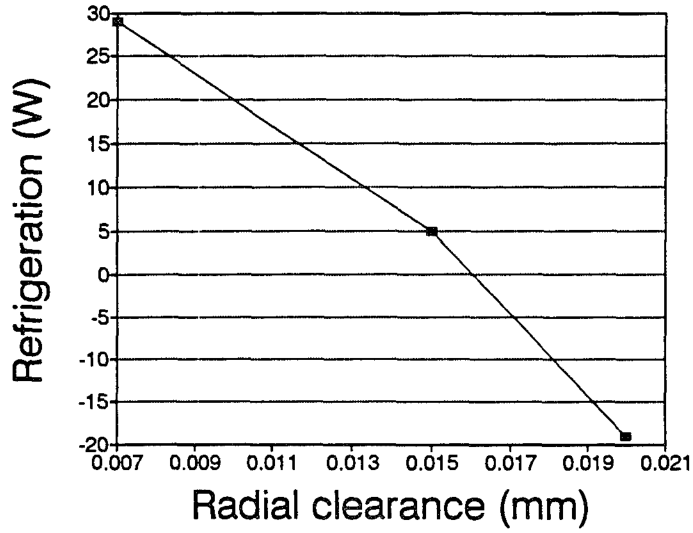



Figure 6.4. Impact of piston/cylinder clearance on refrigeration.

Table 6.3 seems to show that pressure ratio is little affected by changes in clearance between piston and cylinder. The impression is deceptive. The MS*2 code does not permit the user to "lock in" the timing for opening and closing of ports. In the code, the high pressure expansion end ports always open at top dead center. The low pressure compression end ports always close at bottom dead center. The code determines the number of degrees before top dead center that expansion end ports must close in order to bring cylinder pressure to the same level as that of the high pressure ports when they eventually open at top dead center.

Similarly, at the compression end, the MS*2 code determines the number of degrees after bottom dead center that the compression end ports must open to match compression cylinder pressure with the pressure in the ports that eventually open.

The test hardware, on the other hand, was locked into a timing relationship governed by the interaction of ports cut in steel. The timing of the hardware could be altered by adjusting piston position relative to the piston rod, but a change in timing at the expansion end

necessarily involved a corresponding change in the timing of the compression end because the piston was in one piece.

The nominal timing of the piston/cylinder porting arrangements is that the expansion end ports close 30 degrees before top dead center and the compression end ports open 37 degrees past bottom dead center, based upon the pressure-matching concept implicit in the MS*2 code. As is apparent from Table 6.3, that timing would be completely inappropriate if the piston/cylinder clearances are not as assumed.

## 6.2 NITROGEN CONTAMINATION

Both computer predictions and experimental results agree that the tested Sibling design works much better with He than with N2. For example, a version of the second stage running at 10 Hz with mean pressure of 1.5 MPa was predicted to develop 0.52 W of cooling with pure He, 0.39 W with a 50/50 mixture of He and N2 but only 0.2 W on pure N2.

Experimental results were similar. For example, during a series of runs on September 9, 1993, relatively good rates of cooling (up to 0.375 K per minute at the cold end of the second stage regenerator) were obtained with pure He. When N2 was substituted for He, all evidence of cooling vanished and all parts of the second stage began to warm up at an even faster rate.

Although efforts were made to purge N2 (and air) prior to experiments with He, the purging technique was not ideal, and it is probable that the He was to some degree contaminated in many runs that were nominally made with pure He. Better equipment and better technique would eliminate this possible source of reduced performance.

## 6.3 PHENOMENA UNIQUE TO THE SECOND STAGE

The clearance between second stage piston and second stage cylinder head at top dead center is a fraction of a millimeter. The bore of the second stage freezer tube is 1.6 mm, and its volume is small relative to second stage displacement. Movement of the second stage piston forces flow back and forth through the second stage freezer tube even if there is little or no pressure variation in the expansion space of the first stage. If there is no pressure variation in the first stage, fluid simply flows back and forth through the second stage. That flow, which passes through both the second stage heat exchanger and second stage regenerator, causes friction heating losses. Without pressure variation, there is no refrigeration, so the "freezer" warms up.

If the pressure ratio in the first stage is high, the flow in the second stage freezer tube actually decreases. That is because, during expansion, the first stage piston draws fluid out of one end of the second stage regenerator at the same time that the second stage piston is drawing fluid out of the other end. If there were sufficient dead volume in the second stage,

there would be no flow into the second stage regenerator during expansion; all of the expansion would take place in the fluid already there.

There is probably an optimum relationship between pressure change and flow rate in the second stage. It probably lies between the first case (all flow, no pressure change) and the second case (all pressure change, no flow during the expansion stroke). It seems likely that the experiments in this project were conducted much closer to the first case (all flow, no pressure change) than to the optimum. Thus, efforts to increase pressure ratio were probably moving in the right direction.

Unfortunately, practical methods for increasing first stage pressure ratio involved makeshift modifications to the first stage regenerator. That tended to degrade the performance of the first stage, counterbalancing improvements in second stage performance.

## 7.0 SURVEY OF ROTATING/RECIPROCATING MAGNETIC DRIVES

Demonstration of an effective rotating/reciprocating drive would represent a major step forward in the development of this new type of cooler. To date, all experimental work has been done with a relatively crude Sibling Cycle model that uses a gearbox to drive a piston with the requisite rotating/ reciprocating motion. The gearbox has important advantages for experimental work because it ensures positive, mechanical control of piston position at all times. However, the oil-lubricated gearbox is limited in its maximum operating speed and is, in any event, impractical for commercial or space applications.

Part of the Phase I effort was to attempt to identify candidate rotating/reciprocating electromagnetic drive systems for a Sibling compressor/expander. That effort involved consideration of the design requirements and consultation with possible suppliers of candidate drives.

### 7.1 DESIGN REQUIREMENTS

The critical feature of an electromagnetic drive is that the piston is, to some extent, "free". The amplitude of the stroke, and the location in the cylinder where the stroking occurs are determined by the piston's mass and by pressure forces as well as by the forces applied through the electromagnetic drive. Moreover, rotation must be synchronized to insure that cylinder ports open and close in the correct sequence relative to the axial position of the piston.

Available linear drives for Stirling and Brayton equipment are resonant; the pistons and displacers bounce against gas pressure (or mechanical springs) at both ends of their travel. The drive is only required to add energy to the resonant motion; it is not required to overcome large static forces. The Sibling Cycle cooler is well suited to a resonant drive system; the piston runs into a cushion of high pressure fluid at each end of its travel.

Timing of piston rotation relative to reciprocation was a major focus of the experimental work done in Phase I. The test apparatus permitted rotational position of the piston relative to its axial position to be adjusted accurately to any desired relationship. A key finding of the experiments was that timing may vary several degrees from optimum in either direction without significant change in results. That suggests that it should not be difficult to control timing through an electromagnetic drive even though it may not provide as rigid a control of axial piston position as is provided by the mechanical gearbox.

### 7.2 CANDIDATE DRIVE SYSTEMS

Two potential sources of a rotating/reciprocating drives have been identified. One is the existing hardware developed by ADL for use in the Prototype Flight Cooler (PFC) program and now available as surplus. The second is a well-regarded, well-tested linear drive developed by Sunpower, Inc. In the latter case, a rotational drive will have to be imposed upon the linear drive.

7.2.1 The Arthur D. Little, Inc. PFC

The concept of a rotating, reciprocating, ported piston was developed by ADL in a series of Brayton Cycle coolers developed under other contracts over a period of many years ( Refs. 13-16). The last project (the PFC) was concluded in 1990 and the hardware delivered to the Air Force. Among the hardware were several drive systems intended, variously, for preliminary experimental work and for inclusion in the space-qualified unit.

The ADL hardware was attractive because it represented a substantial investment by the United States that had become surplus and was thus available for further use at essentially no cost to the Government. The ADL personnel directly responsible for the overall project and for the drive systems were interviewed. The actual hardware was also examined.

The experimental drives were mechanical units with gears and pulleys. They were nicely designed and built, but bulky. On balance, they did not appear to offer advantages over MS*2's existing gearbox drive and because their ratio of reciprocations is 2:1 rather than the 4:1 required by MS*2's Sibling design, they would have had to be rebuilt to be used. Moreover, they offered no advance, in concept, over MS*2's existing mechanical drive.

In additional to the mechanical drives, ADL had built several electromagnetic rotating/reciprocating drives with integral linear actuators and motors. They were intended to be sealed for extended life in space. The ADL hardware included both a compressor drive of relatively large capacity and an expander drive that was designed to take mechanical power from the first stage expander piston and convert it to electrical power, reclaiming energy from the system. That drive (the "PFC drive") was investigated for possible use in Phase II of this project in close cooperation with the ADL staff members who had originally worked on the PFC project. It is shown in Figure 7.1 (Ref. 15, p. 40).

Although designed as a linear alternator, the actuator (i.e. linear drive component) of the PFC drive would work equally well as a motor. Thus, it could be powered and used to drive the piston of a Sibling Cycle refrigerator. Stroke of the drive is 12.7 mm, which is slightly less than the 15 mm stroke of the testbed apparatus, but would be acceptable with a redesigned piston assembly.

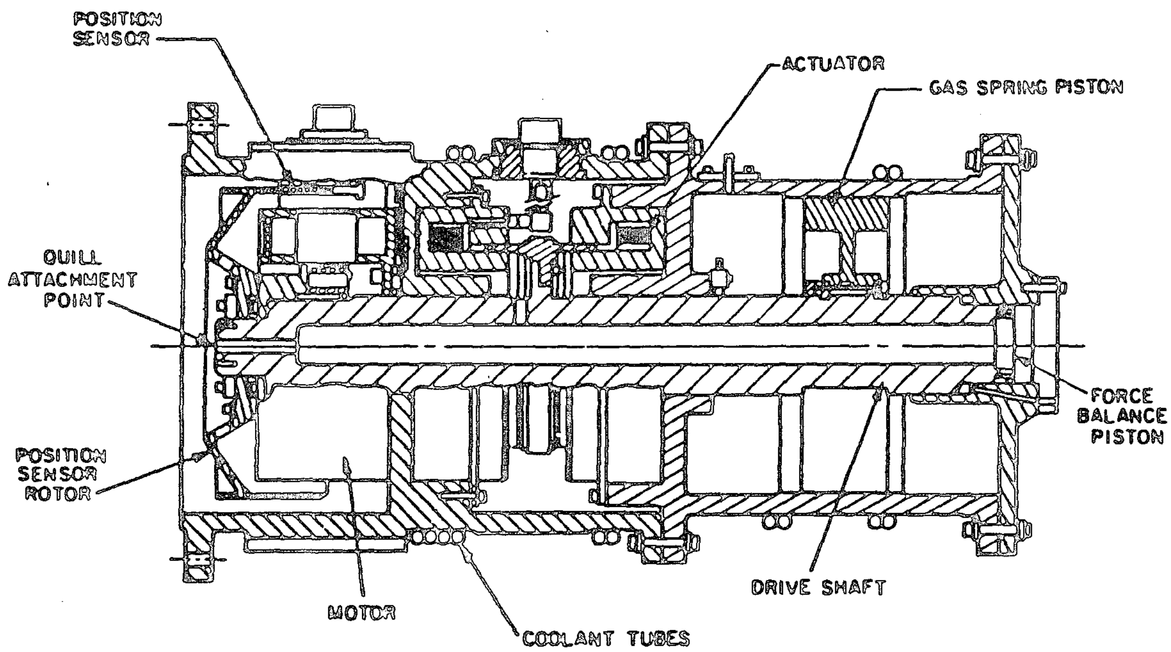



Figure 7.1. The PFC expander drive #1 (from Ref. 15).

The PFC drive actuator was designed for a maximum of 3.73 A at 85 V. It was designed to accept over 60 W of mechanical power, delivering 40 W of electrical power, with a maximum actuator force of 15 kg. This design reflects the relatively low pressures at which the PFC was intended to operate. These electrical parameters might be acceptable in a redesigned Sibling Cycle cooler, but appear low for the existing piston/cylinder arrangements, primarily because the existing Sibling Cycle equipment is designed to operate at relatively higher pressures and mechanical loads.

As originally constructed, the PFC drive was intended to operate in a resonant mode at 40 Hz, with the piston rotating once for every two reciprocations. The rotational speed is thus different from the planned one rotation for every four reciprocations for the Sibling Cycle. Rotational position of the PFC drive is linked to reciprocating position by electronics. The ADL team advised that it would not have been difficult to change the timing circuit to achieve the desired 4/1 ratio of reciprocations to rotations. However, the PFC drive electronics have been partially disassembled and cannibalized for other uses. Control of the drive must therefore be considered problematical.

The most compelling reason for rejecting the PFC drive as an option for further development of the two-stage Sibling Cycle cooler was complexity. The PFC drive was intended to operate for 50,000 hours without maintenance, and to accomplish that feat, it contains a number of extremely complex and delicate features. Many of those features, such as the multiply-redundant position-sensing system, would not be required for a test-bench demonstration of the two-stage Sibling Cycle cooler with electromagnetic rotating/reciprocating drive. Some of those features might be ignored. Others, however, would be likely to create serious problems if the PFC drive were converted for use in Sibling Cycle experiments.

One of the critical subsystems of the PFC drive is the system of gas springs and gas bearings that control the position of the drive shaft. These systems interact; the gas bearings drain to the gas springs. The purpose of the gas springs is to control translational position of the piston so that it does not run into the stops at either end of its allowed travel. Consultations with the ADL project staff reveal that while the gas spring system was made to work in the laboratory tests of the PFC drive, it was still necessary, at the time the PFC project was terminated, to provide multiple, separately regulated gas pressures to different parts of the drive in order to adjust for differential rates of internal flow between gas springs and gas bearings.

Because the logistical problems of running Sibling Cycle experiments are sufficiently difficult by themselves, it was concluded that the additional known problems of controlling the PFC drive might jeopardize the success of a Phase II project predicated on use of the PFC drive, and it was rejected.

### 7.2.2 The Sunpower Drive

The other drive system that was explored for potential use in Phase II is the linear drive developed by Sunpower, Inc., and used successfully in a series of refrigerators operating at temperatures ranging down into the cryogenic region. (Refs 18-21). The Sunpower drive is strictly linear; a separate rotational drive motor would have to be obtained and synchronized with the linear motor.

Dimensions of the drive are suitable for a two-stage Sibling Cycle cryocooler.

The characteristics of the Sunpower drive are favorable. It is more powerful than the drive in ADL's PFC (220 W vs 40 W of pV power) . Part of the difference in power is due to operating speed (60 Hz vs 40 Hz). Part is due to higher efficiency (>90% vs +/- 66%). Mainly, however, it is simply designed to handle more electric current.

Because the actuator of the Sunpower drive is equipped with permanent magnets, it tends to "toggle" if it is rotated. That is probably an advantage for the Sibling Cycle application since it is desirable to cause cylinder wall ports to open and shut abruptly. It would be desirable to arrange the magnets so that the toggle points would "snap" the piston at port opening and closing.

A Sibling Cycle cooler design with stepped piston has been modelled to fit the speed and power parameters of the Sunpower drive. If the proposed cylinder/piston clearances can be achieved, excellent single-stage performance should be attainable. Experience with this contract suggests that the first stage should be fully tested and debugged before the second stage is added.

# 8.0 CONCLUSIONS AND RECOMMENDATIONS

## 8.1 CONCLUSIONS FROM THE EXPERIMENTS PERFORMED

Performance of the second stage built and tested was disappointing. The problem seems to be related in part to a low pressure ratio, which has a particularly adverse effect upon the second stage, where flow unaccompanied by adequate pressure variation produces friction heating that overwhelms the refrigeration effect.

It seems highly probable that leakage through cylinder ports significantly exceeds predictions based upon the MS*2 Stirling Cycle code because the actual piston/cylinder clearance is greater than the clearance that was fed into the code. Subsequent investigation with the MS*2 code has shown that performance is highly sensitive to that clearance.

A closer fit between piston and cylinder would improve performance. That will be difficult to obtain at the dimensions of the existing first stage piston and cylinder, but might be obtained with a smaller piston. To compensate for the smaller displacement, the machine would have to run faster to achieve the same volumetric displacement. The advantage is that both stages could be smaller without loss of capacity.

So far, the only information about what has been going on in the second stage has come from thermocouples attached to the outer surfaces of second stage components and from calculations based on pV data from the first stage expansion space. It would be desirable to tap into the freezer tube of the second stage with a pressure transducer so that the actual pV diagram of the second stage could be constructed from direct measurements.

## 8.2 RECOMMENDATIONS FOR FUTURE WORK

Based upon the conclusions above, performance of a two-stage Sibling Cycle refrigerator would be greatly improved if leakage at the cylinder ports could be reduced. That could be accomplished by making the piston/cylinder clearance tighter and more consistent and by running the machine faster (to decrease leak time and improve the hydrodynamic gas bearing). The piston/cylinder clearance could be tighter if the parts were smaller.

With a longer ratio of piston length to diameter (l/d ratio), the effectiveness of the hydrodynamic gas bearing could be improved, thereby tending to center the piston better and maintain proper clearance all around it. Figure 8.1 is a graph illustrating the relationship between l/d ratio and bearing capacity at varying eccentricities.

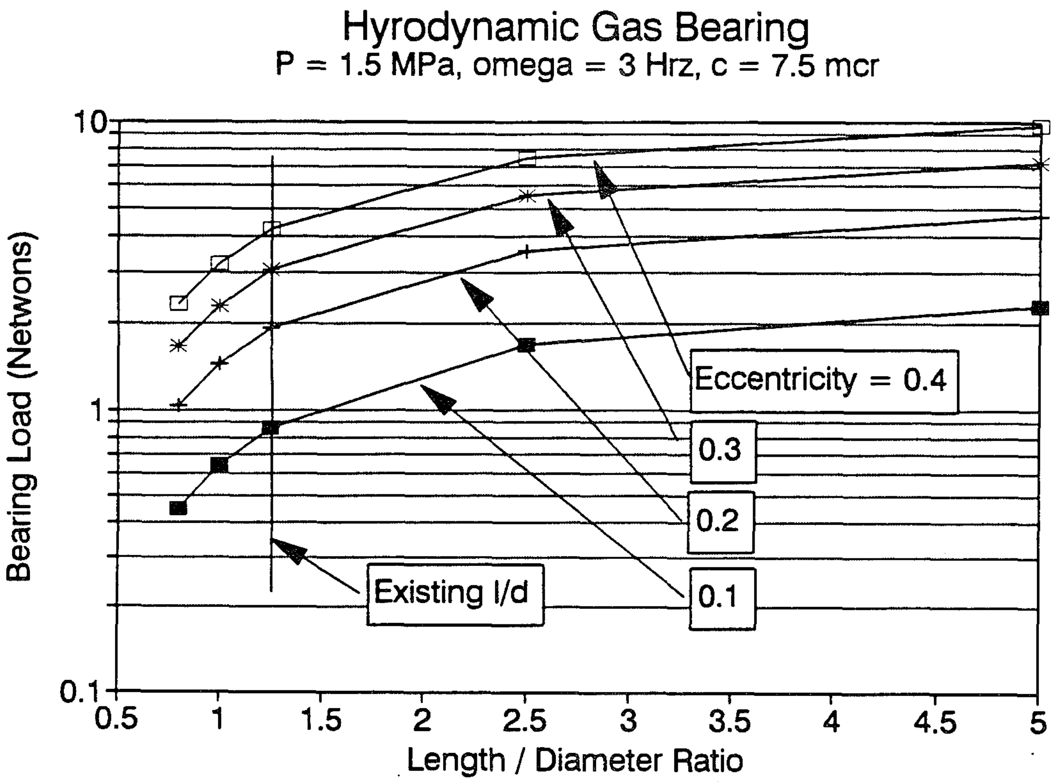



Figure 8.1. Effect of l/d ratio on bearing capacity.

The Sunpower linear drive could be adapted to run a newly designed two-stage Sibling Cycle cryocooler at 60 Hz. That speed is about six times higher than the apparently optimal operating speed of the existing unit. The load capacity of the hydrodynamic gas bearing improves at higher rotational speeds (Ref. 17). At 60 Hz, a smaller piston (with a tighter clearance and more effective hydrodynamic gas bearing) would be practical. A promising design of the first stage compression/expansion unit for a new 60 Hz Sibling has been modelled with the MS*2 Stirling Cycle Code.

A well-designed Sibling Cycle cryocooler continues to hold out the promise of a simple, compact, robust and reliable cryocooler with a single moving part.

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# APPENDIX A

## SELECTED RESULTS

## MS*2 CODE

## SECOND STAGE

MITCHELL/STIRLING MACHINES/SYSTEMS, INC.

Sibling Cycle cooler - second stage - MS*2 Stirling Cycle Code

Unvarying parameters:

Bore = 12.7 mm
Stroke = 15.0 mm
Regenerator fill = .28
Time steps = 1,440
Space steps = 154

Regenerator diameter = 15.875 mm
Regenerator material = stainless steel
Regenerator spaces = 126

| File | Temperatures | | Speed | Fluid | Mean P | Reg L | Tube L | Wire D | Duct V (cm^3) | | Viscous | P | Port time | | Reject | Refrig |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| name | reject | absorb | (Hz) | | (MPa) | (mm) | (mm) | (mm) | Exp | Comp | Loss (W) | ratio | Exp | Comp | (W) | (W) |
| TQ1-6-13 | 250 | 100 | 16.67 | He | 4 | 60 | 80 | 0.04 | 0.02 | 6.9 | 3.54 | 1.27 | 39 | 41.5 | 24.151 | 1.79 |
| TQ2-6-13 | 250 | 100 | 16.67 | He | 4 | 60 | 80 | 0.04 | 0.02 | 9.9 | 2.28 | 1.22 | 31 | 39.5 | 14.07 | 0.175 |
| TQ1-6-14 | 250 | 100 | 16.67 | He | 4 | 60 | 80 | 0.04 | 0.02 | 6.9 | 2.53 | 1.27 | 39 | 37 | 19.48 | 1.765 |
| TQ5-6-25 | 250 | 100 | 16.67 | He | 4 | 60 | 100 | 0.025 | 0.02 | 9.9 | 3.75 | 1.23 | 29 | 39.5 | 15.28 | 0.029 |
| TQ1-6-26 | 250 | 100 | 16.67 | He | 4 | 60 | 100 | 0.025 | 0.06 | 12.87 | 3.69 | 1.2 | 29.8 | 38 | 13.915 | -0.478 |
| TQ2-6-26 | 250 | 100 | 15 | He | 4 | 60 | 100 | 0.025 | 0.06 | 12.87 | 2.84 | 1.2 | 29.8 | 38 | 12.36 | -0.067 |
| TQ3-6-26 | 250 | 100 | 13.33 | He | 4 | 60 | 100 | 0.025 | 0.06 | 12.87 | 2.13 | 1.2 | 29.8 | 38 | 10.9 | 0.246 |
| TQ4-6-26 | 250 | 100 | 11.67 | He | 4 | 60 | 100 | 0.025 | 0.06 | 12.87 | 1.54 | 1.2 | 29.8 | 38 | 9.46 | 0.465 |
| TQ1-8-1 | 300 | 290 | 10 | He | 1.5 | 60 | 100 | 0.04 | 0.06 | 23.75 | 0.37 | 1.11 | 31.3 | 34 | 1.7 | 0.739 |
| TQ2-8-1 | 300 | 290 | 10 | He | 1.5 | 60 | 100 | 0.025 | 0.06 | 23.75 | 0.76 | 1.11 | 31.3 | 34 | 1.78 | 0.44 |
| TQ3-8-1 | 300 | 290 | 10 | He | 1.5 | 60 | 100 | 0.05 | 0.06 | 23.75 | 0.29 | 1.11 | 31.3 | 34 | 1.68 | 0.81 |
| TQ1-8-2 | 300 | 290 | 10 | He | 1.5 | 50 | 100 | 0.04 | 0.06 | 23.75 | 0.33 | 1.11 | 32 | 35 | 1.76 | 0.782 |
| TQ2-8-2 | 300 | 290 | 10 | He | 1.5 | 48 | 100 | 0.04 | 0.06 | 23.75 | 0.32 | 1.12 | 32 | 35.3 | 1.77 | 0.799 |
| TQ3-8-2 | 300 | 290 | 10 | He | 1.5 | 60 | 100 | 0.025 | 0.06 | 23.75 | 0.64 | 1.11 | 32 | 35 | 1.82 | 0.521 |
| TQ4-8-2 | 300 | 290 | 10 | N2 | 1.5 | 50 | 100 | 0.025 | 0.06 | 23.75 | 1.05 | 1.11 | 34 | 36.8 | 1.64 | 0.204 |
| TQ5-8-2 | 300 | 290 | 10 | N2/He | 1.5 | 50 | 100 | 0.025 | 0.06 | 23.75 | 0.85 | 1.11 | 33.5 | 36.3 | 1.72 | 0.388 |
| TQ1-8-3 | 300 | 290 | 10 | N2 | 1.5 | 60 | 100 | 0.05 | 0.06 | 23.75 | 1.19 | 1.11 | 31.3 | 34 | 1.63 | 0.089 |
| TQ2-8-3 | 300 | 290 | 10 | N2/He | 1.5 | 60 | 100 | 0.05 | 0.06 | 23.75 | 0.98 | 1.11 | 32.8 | 35.3 | 1.69 | 0.279 |
| TQ1-8-4 | 300 | 290 | 13.33 | N2 | 1.5 | 60 | 100 | 0.025 | 0.06 | 23.75 | 2.41 | 1.11 | 33.5 | 35.8 | 2.32 | -0.516 |
| TQ2-8-4 | 300 | 290 | 13.33 | N2/He | 1.5 | 60 | 100 | 0.05 | 0.06 | 23.75 | 1.92 | 1.11 | 32.8 | 35.3 | 2.381 | -0.094 |
| TQ1-8-5 | 300 | 290 | 10 | He | 1.5 | 50 | 100 | 0.05 | 0.06 | 23.75 | 0.25 | 1.11 | 32 | 35 | 1.74 | 0.853 |
| TQ2-8-5 | 300 | 290 | 10 | He | 1.5 | 50 | 100 | 0.06 | 0.06 | 23.75 | 0.21 | 1.11 | 32 | 35 | 1.74 | 0.882 |
| TQ3-8-5 | 300 | 290 | 10 | He | 1.5 | 50 | 100 | 0.08 | 0.06 | 23.75 | 0.17 | 1.12 | 31.8 | 35 | 1.73 | 0.912 |
| TQ4-8-5 | 300 | 290 | 10 | N2 | 1.5 | 50 | 100 | 0.08 | 0.06 | 23.75 | 0.53 | 1.11 | 34 | 36.8 | 1.51 | 0.563 |
| TQ5-8-5 | 300 | 290 | 10 | N2 | 1.5 | 50 | 100 | 0.1 | 0.06 | 23.75 | 0.51 | 1.11 | 34 | 36.8 | 1.5 | 0.554 |
| TQ1-8-23 | 300 | 290 | 10 | He | 1.5 | 60 | 120 | 0.04 | 0.06 | 23.75 | 0.38 | 1.11 | 31.3 | 34 | 1.7 | 0.734 |
| TQ2-8-23 | 300 | 290 | 10 | He | 1.5 | 60 | 140 | 0.04 | 0.06 | 23.75 | 0.39 | 1.11 | 31.3 | 34 | 1.7 | 0.725 |

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# APPENDIX B

## SUMMARY OF HARDWARE

## MODIFICATIONS AND TEST RESULTS

MITCHELL/STIRLING MACHINES/SYSTEMS, INC.
Sibling Cycle cooler - two stage experiments

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| PISTONS | 1ST STAGE | 2D STAGE |
|---|---|---|
| Stroke (mm) | 15 | 15 |
| Bore (mm) | 42 | 12.7 |

TIMING

| | |
|---|---|
| Expansion end | 30.5 degrees before TDC |
| Compression end | 37.5 degrees after BDC |

| Date | Run | Pmax | Speed | Gas | Tim'g | Config'n | | P | Time | Cooling Rate (C/min) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | # | (MPa) | (Hz) | | (Rel.) | 1st | 2nd | Ratio | (s) | ThrmA | ThrmB | ThrmC | ThrmD | ThrmE | ThrmF | ThrmG | ThrmH | ThrmI |
| 7/12/93 | 3 | | 7.8 | He | 40 | A | A | 1.11 | 570 | 0.08 | | | | | | | | |
| " | 4 | | 9.7 | " | 40 | " | " | 1.12 | 852 | 0.08 | | | | | | | | |
| 7/15/93 | 1 | | 8.5 | " | 30 | B | B | | 265 | 0.34 | | | | | | | | |
| " | 2 | | " | " | 25 | " | " | | 145 | 0.17 | | | | | | | | |
| " | 3 | | " | " | 29 | " | " | | 175 | 0.31 | | | | | | | | |
| " | 6 | | " | " | 31 | " | A | 1.16 | 670 | 0.125 | | | | | | | | |
| " | 7 | | " | " | " | " | " | 1.16 | 255 | 0.118 | | | | | | | | |
| 7/19/93 | 13 | | | " | 45 | C | " | 1.16 | 99 | 0.12 | | | | | | | | |
| 7/20/93 | 2 | 2.25 | 8.5 | " | 43 | " | " | 1.15 | | | | | | | | | | |
| " | 3 | 2.25 | " | " | 41 | " | " | 1.14 | | | | | | | | | | |
| 7/24/93 | 1 | | " | " | 45 | " | " | 1.15 | | | | | | | | | | |
| " | 3 | | " | " | 44 | " | " | 1.18 | | | | | | | | | | |
| 7/27/93 | 1 | 2.31 | " | " | 43 | " | B | 1.2 | 180 | 0.33 | | | | | | | | |
| " | 2 | 1.98 | " | " | " | " | " | 1.2 | 385 | 0.31 | | | | | | | | |
| " | 3 | 1.21 | 9 | " | " | " | " | 1.21 | 305 | 0.3 | | | | | | | | |
| " | 4 | 1.49 | " | " | " | " | " | 1.19 | 260 | 0.16 | | | | | | | | |
| " | 5 | 1.8 | 8.7 | " | " | " | " | 1.2 | 180 | 0.2 | | | | | | | | |
| " | 6 | " | 5.5 | " | " | " | " | 1.19 | 135 | -0.1 | | | | | | | | |
| " | 7 | " | 11 | " | " | " | " | 1.2 | 235 | 0.36 | | | | | | | | |
| " | 8 | " | 9 | " | " | " | " | 1.2 | 200 | 0.27 | | | | | | | | 0.24 |
| " | 9 | " | 9 | " | " | " | " | 1.19 | 240 | 0.2 | | | | | | | | 0.125 |
| 7/28/93 | 1 | | 9 | " | 45 | " | A | | 255 | | | -0.24 | | | | | | |
| " | 2 | | " | " | " | " | " | 1.19 | 65 | | | -0.18 | | | | | | |
| " | 3 | | " | " | " | " | " | 1.19 | 355 | | | -0.16 | | | | | | |
| " | 4 | | 6 | " | " | " | " | 1.17 | 125 | | | -0.15 | | | | | | |

| CONFIGURATIONS | First stage | Second stage | Thermocouple placement |
|---|---|---|---|
| | A = original low-lift design | A = Original 2d stage | A = 2d stage freezer tube |
| | B = Full size regen, Pb shot | B = 1 m Cu tube, 1.6 mm ID | B = Base of copper regen tube |
| | C = "B" regen, balls in ducts | C = 10mm Pb shot, 50mm scrn | C = 2d stage freezer tube near regen |
| | D = Small bore regen, empty | D = .5 m Cu tube, 1.6 mm ID | D = 2d stage freezer tube near cylinder |
| | E = "A" regen, balls in ducts | E = 30mm Pb shot, 30mm scrn | E = Warm end, 2d stg regen |
| | F = small regen, ducts open | F = 30mm ss bb, 30mm scrn | F = Cold end, 2d stg regen |
| | G = new plug, bypass tubes | G = 1.6 & 3.2mm ss bb | G = Expansion end, 2d stg cyl |
| | | H = G10 plug + 100 scrns | H = Base, 2d stg cylinder |
| | | | I = 1st stg cylinder head |

MITCHELL/STIRLING MACHINES/SYSTEMS, INC.
Sibling Cycle cooler - two stage experiments

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| PISTONS | 1ST STAGE | 2D STAGE |
|---|---|---|
| Stroke (mm) | 15 | 15 |
| Bore (mm) | 42 | 12.7 |

TIMING

| | |
|---|---|
| Expansion end | 30.5 degrees before TDC |
| Compression end | 37.5 degrees after BDC |

| Date | Run | Pmax | Speed | Gas | Tim'g | Config'n | | P | Time | Cooling Rate (C/min) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | # | (MPa) | (Hz) | | (Rel.) | 1st | 2nd | Ratio | (s) | ThrmA | ThrmB | ThrmC | ThrmD | ThrmE | ThrmF | ThrmG | ThrmH | ThrmI |
| 7/28/93 | 5 | 2.21 | 10.5 | He | 45 | C | A | 1.18 | 120 | | | -0.15 | | | | | | |
| " | 6 | 2.21 | 12 | " | " | " | " | 1.19 | 120 | | | " | | | | | | |
| " | 7 | 2.22 | 9 | " | " | " | " | 1.18 | 420 | | | -0.16 | | | | | | |
| " | 8 | 2.98 | " | " | " | " | " | " | 95 | | | -0.13 | | | | | | |
| " | 9 | 1.62 | 10.5 | " | " | " | " | 1.19 | 210 | | | -0.26 | | | | | | |
| " | 10 | 1.59 | " | " | " | " | " | 1.2 | 180 | | | " | | | | | | |
| " | 11 | | 10 | " | " | " | " | | 120 | | | " | | | | | | |
| " | 12 | 1.7 | 8.5 | " | " | " | C | 1.2 | 180 | | | -0.23 | | | | | | |
| " | 13 | 1.64 | 10.5 | " | " | | " | 1.2 | " | | | -0.1 | | | | | | |
| 7/30/93 | 2 | 1.71 | 9 | " | " | " | " | 1.18 | 140 | | | -0.39 | | | | | | |
| " | 3 | " | " | " | " | " | " | 1.19 | 240 | | | -0.2 | | | | | | |
| " | 4 | " | " | " | " | " | " | " | 300 | | | -0.14 | | | | | | |
| " | 5 | 1.72 | " | " | " | " | " | " | " | | | -0.16 | | | | | | |
| 7/31/93 | 6 | 0.8 | 8 | " | -15 | D | " | 1.27 | 70 | -1.37 | | | | | | | | |
| " | 8 | 2.94 | 8.5 | " | " | " | " | 1.2 | 125 | -1.73 | | | | | | | | |
| " | 9 | 2.2 | 5.5 | " | " | " | " | 1.22 | 50 | -2.04 | | | | | | | | |
| " | 11 | 1.52 | 9 | " | -10 | " | " | 1.25 | 75 | -1.68 | | | | | | | | |
| " | 12 | 0.92 | " | " | -12 | " | " | " | 185 | -0.71 | | | | | | | | |
| 8/13/93 | 7 | 2.05 | 8.5 | " | 40 | E | " | 1.12 | 300 | | | | -0.06 | | | | | |
| " | 8 | 1.85 | " | " | " | " | " | " | " | | | | " | | | | | |
| " | 9 | 1.44 | 10 | " | " | " | " | " | " | | | | -0.08 | | | | | |
| " | 10 | 1.6 | 7 | " | " | " | " | " | " | | | | 0.02 | | | | | |
| " | 11 | 1.57 | 5 | " | " | " | " | 1.106 | " | | | | 0 | | | | | |
| " | 12 | 1.6 | 12 | " | " | " | " | " | " | | | | -0.2 | | | | | |

| CONFIGURATIONS | First stage | Second stage | Thermocouple placement |
|---|---|---|---|
| | A = original low-lift design | A = Original 2d stage | A = 2d stage freezer tube |
| | B = Full size regen, Pb shot | B = 1 m Cu tube, 1.6 mm ID | B = Base of copper regen tube |
| | C = "B" regen, balls in ducts | C = 10mm Pb shot, 50mm scrn | C = 2d stage freezer tube near regen |
| | D = Small bore regen, empty | D = .5 m Cu tube, 1.6 mm ID | D = 2d stage freezer tube near cylinder |
| | E = "A" regen, balls in ducts | E = 30mm Pb shot, 30mm scrn | E = Warm end, 2d stg regen |
| | F = small regen, ducts open | F = 30mm ss bb, 30mm scrn | F = Cold end, 2d stg regen |
| | G = new plug, bypass tubes | G = 1.6 & 3.2mm ss bb | G = Expansion end, 2d stg cyl |
| | | H = G10 plug + 100 scrns | H = Base, 2d stg cylinder |
| | | | I = 1st stg cylinder head |

| PISTONS | 1ST STAGE | 2D STAGE |
|---|---|---|
| Stroke (mm) | 15 | 15 |
| Bore (mm) | 42 | 12.7 |

| TIMING | |
|---|---|
| Expansion end | 30.5 degrees before TDC |
| Compression end | 37.5 degrees after BDC |

| Date | Run | Pmax | Speed | Gas | Tim'g | Config'n | | P | Time | Cooling Rate (C/min) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | # | (MPa) | (Hz) | | (Rel.) | 1st | 2nd | Ratio | (s) | ThrmA | ThrmB | ThrmC | ThrmD | ThrmE | ThrmF | ThrmG | ThrmH | ThrmI |
| 8/13/93 | 13 | 2.61 | 7 | He | 40 | E | C | 1.12 | 600 | | | | 0.06 | | | | | |
| " | 14 | 1.56 | " | " | " | " | " | 1.11 | 300 | | | | 0 | | | | | |
| " | 15 | 1.87 | " | " | 45 | " | " | 1.1 | | | | | | | | | | |
| " | 16 | 1.54 | " | " | 35 | " | " | " | 360 | | | | 0.066 | | | | | |
| " | 17 | 1.21 | " | " | " | " | " | 1.11 | 300 | | | | 0 | | | | | |
| " | 18 | 1.4 | " | " | 33 | " | " | " | 120 | | | | -0.25 | | | | | |
| " | 19 | " | " | " | 37 | " | " | 1.12 | 360 | | | | 0 | | | | | |
| " | 20 | " | " | " | 39 | " | " | 1.11 | 180 | | | | -0.1 | | | | | |
| " | 21 | 1.39 | " | " | " | " | " | 1.12 | 240 | | | | -0.05 | | | | | |
| 8/17/93 | 2 | 2.11 | 9 | " | 40 | " | D | 1.13 | 200 | | | | 0.45 | | | | | |
| " | 3 | " | " | " | " | " | " | 1.12 | 120 | | | | 0.1 | | | | | |
| " | 4 | 2.18 | " | " | 38 | " | " | " | 180 | | | | 0.5 | | | | | |
| " | 5 | 2.15 | " | " | " | " | " | 1.13 | 300 | | | | 0.24 | | | | | |
| " | 6 | 1.84 | " | " | " | " | " | 1.12 | " | | | | 0.12 | | | | | |
| " | 7 | 1.49 | " | " | " | " | " | " | " | | | | -0.1 | | | | | |
| " | 8 | 2.33 | " | " | " | " | " | 1.13 | " | | | | 0.16 | | | | | |
| " | 9 | 2.31 | " | " | 36 | " | " | 1.121 | " | | | | 0.14 | | | | | |
| " | 10 | 2.28 | 7 | " | " | " | " | 1.118 | " | | | | -0.06 | | | | | |
| " | 11 | " | 10.5 | " | " | " | " | 1.123 | 120 | | | | 0.2 | | | | | |
| 8/18/93 | 3 | 1.79 | 9 | " | 38 | " | E | 1.12 | 300 | 0.1 | | | | | | | | |
| " | 5 | 2.09 | " | " | " | " | " | " | " | 0.08 | | | | | | | | |
| " | 7 | 1.93 | " | " | 40 | " | " | " | " | 0.02 | | | | | | | | |
| " | 9 | 1.95 | " | " | 36 | " | " | 1.11 | " | 0.06 | | | | | | | | |
| 8/19/93 | 1 | 1.75 | 8.5 | " | 38 | " | " | 1.115 | " | 0.12 | | | | | | | | |
| " | 2 | 1.73 | 7 | " | " | " | " | 1.11 | " | 0.04 | | | | | | | | |

| CONFIGURATIONS | First stage | Second stage | Thermocouple placement |
|---|---|---|---|
| | A = original low-lift design | A = Original 2d stage | A = 2d stage freezer tube |
| | B = Full size regen, Pb shot | B = 1 m Cu tube, 1.6 mm ID | B = Base of copper regen tube |
| | C = "B" regen, balls in ducts | C = 10mm Pb shot, 50mm scrn | C = 2d stage freezer tube near regen |
| | D = Small bore regen, empty | D = .5 m Cu tube, 1.6 mm ID | D = 2d stage freezer tube near cylinder |
| | E = "A" regen, balls in ducts | E = 30mm Pb shot, 30mm scrn | E = Warm end, 2d stg regen |
| | F = small regen, ducts open | F = 30mm ss bb, 30mm scrn | F = Cold end, 2d stg regen |
| | G = new plug, bypass tubes | G = 1.6 & 3.2mm ss bb | G = Expansion end, 2d stg cyl |
| | | H = G10 plug + 100 scrns | H = Base, 2d stg cylinder |
| | | | I = 1st stg cylinder head |

MITCHELL/STIRLING MACHINES/SYSTEMS, INC.
Sibling Cycle cooler - two stage experiments

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| PISTONS | 1ST STAGE | 2D STAGE |
|---|---|---|
| Stroke (mm) | 15 | 15 |
| Bore (mm) | 42 | 12.7 |

| TIMING | |
|---|---|
| Expansion end | 30.5 degrees before TDC |
| Compression end | 37.5 degrees after BDC |

| Date | Run | Pmax | Speed | Gas | Tim'g | Config'n | | P | Time | Cooling Rate (C/min) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | # | (MPa) | (Hz) | | (Rel.) | 1st | 2nd | Ratio | (s) | ThrmA | ThrmB | ThrmC | ThrmD | ThrmE | ThrmF | ThrmG | ThrmH | ThrmI |
| 8/19/93 | 3 | 1.75 | 10.5 | He | 38 | E | E | 1.115 | 300 | -0.04 | | | | | | | | |
| " | 4 | 1.73 | 8.5 | " | " | " | " | 1.116 | " | 0.02 | | | | | | | | |
| " | 5 | 1.74 | 10.5 | " | " | " | " | 1.115 | " | 0.08 | | | | | | | | |
| " | 6 | 1.73 | 7 | " | " | " | " | 1.116 | " | -0.02 | | | | | | | | |
| " | 7 | 1.74 | 8.5 | " | " | " | " | 1.115 | " | -0.02 | | | | | | | | |
| " | 8 | 2.95 | " | " | " | " | " | 1.12 | 120 | -0.15 | | | | | | | | |
| " | 10 | 1.68 | " | " | " | " | F | 1.11 | 300 | 0.08 | | | | | | | | |
| " | 11 | " | " | " | " | " | " | " | " | 0.1 | | | | | | | | |
| " | 12 | " | " | " | " | " | " | " | " | 0.06 | | | | | | | | |
| " | 13 | 2 | " | " | " | " | " | " | " | 0.16 | | | | | | | | |
| " | 14 | " | " | " | " | " | " | | " | 0.1 | | | | | | | | |
| 8/20/93 | 1 | 1.69 | 9 | " | | " | G | 1.11 | " | 0.34 | | | | | | | | |
| " | 2 | " | " | " | | " | " | " | 240 | 0.125 | | | | | | | | |
| " | 3 | | 10.5 | " | | " | " | " | 360 | 0.1 | | | | | | | | |
| " | 4 | 1.65 | 7 | " | | " | " | " | 300 | -0.06 | | | | | | | | |
| " | 5 | 1.67 | 10.5 | " | | " | " | 1.12 | " | 0.04 | | | | | | | | |
| 8/26/93 | 2 | 1.38 | 9 | " | 36 | F | " | 1.2 | 285 | 0.08 | | | | | | | | |
| " | 3 | 1.02 | " | " | 37 | " | " | 1.21 | 300 | -0.04 | | | | | | | | |
| " | 4 | 1.92 | " | " | " | " | " | " | 240 | 0 | | | | | | | | |
| 8/27/93 | 1 | 1.36 | " | " | " | " | " | 1.2 | 210 | | | 0.14 | 0.11 | 0 | 0.23 | 0 | 0 | |
| " | 2 | | 12.5 | " | " | " | " | | 600 | | | 0.26 | 0.04 | -0.07 | 0.29 | 0 | 0 | |
| " | 4 | 1.31 | 9 | " | " | " | " | 1.21 | 300 | | | 0.12 | 0.08 | 0.04 | 0.12 | 0 | -0.1 | |
| " | 6 | 2.22 | " | " | " | " | " | " | " | | | 0.1 | | 0 | 0.1 | 0 | 0 | |
| " | 7 | 1.62 | " | " | 38 | " | " | | " | | | 0.14 | | 0 | -0.4 | | 0 | |
| " | 8 | | " | " | 39 | " | " | | 400 | | | | | 0.12 | | | | |

| CONFIGURATIONS | First stage | Second stage | Thermocouple placement |
|---|---|---|---|
| | A = original low-lift design | A = Original 2d stage | A = 2d stage freezer tube |
| | B = Full size regen, Pb shot | B = 1 m Cu tube, 1.6 mm ID | B = Base of copper regen tube |
| | C = "B" regen, balls in ducts | C = 10mm Pb shot, 50mm scrn | C = 2d stage freezer tube near regen |
| | D = Small bore regen, empty | D = .5 m Cu tube, 1.6 mm ID | D = 2d stage freezer tube near cylinder |
| | E = "A" regen, balls in ducts | E = 30mm Pb shot, 30mm scrn | E = Warm end, 2d stg regen |
| | F = small regen, ducts open | F = 30mm ss bb, 30mm scrn | F = Cold end, 2d stg regen |
| | G = new plug, bypass tubes | G = 1.6 & 3.2mm ss bb | G = Expansion end, 2d stg cyl |
| | | H = G10 plug + 100 scrns | H = Base, 2d stg cylinder |
| | | | I = 1st stg cylinder head |

MITCHELL/STIRLING MACHINES/SYSTEMS, INC.
Sibling Cycle cooler - two stage experiments



| PISTONS | 1ST STAGE | 2D STAGE |
|---|---|---|
| Stroke (mm) | 15 | 15 |
| Bore (mm) | 42 | 12.7 |

| TIMING | |
|---|---|
| Expansion end | 30.5 degrees before TDC |
| Compression end | 37.5 degrees after BDC |

| Date | Run | Pmax | Speed | Gas | Tim'g | Config'n | | P | Time | Cooling Rate (C/min) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | # | (MPa) | (Hz) | | (Rel.) | 1st | 2nd | Ratio | (s) | ThrmA | ThrmB | ThrmC | ThrmD | ThrmE | ThrmF | ThrmG | ThrmH | ThrmI |
| 8/27/93 | 11 | | 9 | He | 39 | F | G | | 200 | | | 0.514 | | -0.17 | | | | |
| " | 13 | 0.8 | " | " | " | " | " | 1.25 | | | | | | | | | | |
| 8/30/93 | 3 | | " | " | " | " | " | | 180 | | | 0.13 | 0 | | | 1.1 | 0.233 | |
| " | 4 | | " | " | 36 | " | " | | 240 | | | 0 | 0.325 | | | 0.425 | -0.06 | |
| 9/9/93 | 4 | 1.8 | " | " | 39 | " | H | 1.21 | 270 | | | 0.24 | 0.186 | | 0.375 | 0.361 | | |
| " | 6a | 2 | " | " | " | " | " | " | " | | | 0.11 | 0.24 | | 0.29 | 0.31 | | |
| " | 6b | " | " | " | " | " | " | " | " | | | 0.3 | 0.1 | | 0.3 | 0.3 | | |
| " | 9 | " | " | N2 | " | " | " | 1.25 | 300 | | | -0.44 | -0.48 | | -0.4 | -0.24 | | |
| 10/21/93 | 5 | 1.32 | 8.5 | He | 40 | G | " | 1.2 | 75 | -0.32 | | | | | | | | |
| " | 6 | 1.21 | " | " | 45 | " | " | 1.21 | 70 | 0.17 | | | | | | | | |
| " | 7 | 1.17 | " | " | 43 | " | " | 1.26 | 80 | 0 | | | | | | | | |
| " | 8 | 1.13 | 7 | " | " | " | " | 1.27 | 30 | | | | | | | | | |
| " | 9 | 1.2 | 10.5 | " | " | " | " | 1.28 | 50 | | | | | | | | | |
| " | 10 | 1.19 | 8.5 | " | " | " | " | 1.27 | 45 | | | | | | | | | |
| " | 11 | 1.19 | 12 | " | " | " | " | " | 95 | | | | | | | | | |
| " | 12 | 0.82 | 8.5 | " | " | " | " | 1.3 | 105 | -0.57 | | | | | | | | |
| " | 13 | 0.57 | 12 | " | " | " | " | " | 145 | -0.46 | | | | | | | | |
| " | 15 | 0.25 | 14 | " | " | " | " | 1.48 | 125 | | | | | | | | | |
| " | 16 | 1.38 | 8.5 | " | " | " | " | 1.2 | 25 | | | | | | | | | |
| " | 17 | 1.52 | 12 | " | " | " | " | 1.24 | 90 | | | | | | | | | |
| " | 18 | 1.42 | 7 | " | " | " | " | " | 45 | | | | | | | | | |
| " | 19 | 0.97 | 8.5 | " | " | " | " | 1.2 | 30 | | | | | | | | | |
| " | 20 | 0.55 | 14 | " | " | " | " | 1.19 | 90 | -0.06 | | | | | | | | |
| 10/26/93 | 7 | 0.95 | 11.5 | " | " | " | " | 1.26 | 300 | 0.08 | | 0.06 | 0 | | 0.04 | 0.5 | | |
| " | 10 | " | " | " | " | " | " | " | " | 0.11 | | 0 | 0.06 | | 0.04 | 0.48 | | |

| CONFIGURATIONS | First stage | Second stage | Thermocouple placement |
|---|---|---|---|
| | A = original low-lift design | A = Original 2d stage | A = 2d stage freezer tube |
| | B = Full size regen, Pb shot | B = 1 m Cu tube, 1.6 mm ID | B = Base of copper regen tube |
| | C = "B" regen, balls in ducts | C = 10mm Pb shot, 50mm scrn | C = 2d stage freezer tube near regen |
| | D = Small bore regen, empty | D = .5 m Cu tube, 1.6 mm ID | D = 2d stage freezer tube near cylinder |
| | E = "A" regen, balls in ducts | E = 30mm Pb shot, 30mm scrn | E = Warm end, 2d stg regen |
| | F = small regen, ducts open | F = 30mm ss bb, 30mm scrn | F = Cold end, 2d stg regen |
| | G = new plug, bypass tubes | G = 1.6 & 3.2mm ss bb | G = Expansion end, 2d stg cyl |
| | | H = G10 plug + 100 scrns | H = Base, 2d stg cylinder |
| | | | I = 1st stg cylinder head |

| PISTONS | 1ST STAGE | 2D STAGE |
|---|---|---|
| Stroke (mm) | 15 | 15 |
| Bore (mm) | 42 | 12.7 |

TIMING

| | |
|---|---|
| Expansion end | 30.5 degrees before TDC |
| Compression end | 37.5 degrees after BDC |

| Date | Run | Pmax | Speed | Gas | Tim'g | Config'n | | P | Time | Cooling Rate (C/min) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | # | (MPa) | (Hz) | | (Rel.) | 1st | 2nd | Ratio | (s) | ThrmA | ThrmB | ThrmC | ThrmD | ThrmE | ThrmF | ThrmG | ThrmH | ThrmI |
| 10/28/93 | 4 | 0.77 | 18.6 | He | 43 | G | H | 1.3 | 300 | -0.48 | | -0.42 | -0.72 | | -0.22 | -0.28 | | |
| " | 6 | 0.71 | 8.7 | " | " | " | " | 1.28 | " | -0.04 | | -0.08 | 0 | | 0 | 0.18 | | |
| " | 8 | 0.75 | 4.9 | " | " | " | " | 1.24 | " | -0.1 | | -0.06 | -0.02 | | -0.02 | -0.04 | | |
| " | 10 | 0.8 | 10.7 | " | " | " | " | 1.27 | " | -0.24 | | -0.24 | -0.2 | | -0.04 | 0.2 | | |
| " | 12 | 0.7 | 6.8 | " | " | " | " | 1.285 | " | 0 | | -0.1 | 0 | | -0.02 | 0.08 | | |
| " | 14 | 0.5 | 8.7 | " | " | " | " | | " | -0.08 | | -0.14 | 0 | | 0 | 0.02 | | |

| CONFIGURATIONS | First stage | Second stage | Thermocouple placement |
|---|---|---|---|
| | A = original low-lift design | A = Original 2d stage | A = 2d stage freezer tube |
| | B = Full size regen, Pb shot | B = 1 m Cu tube, 1.6 mm ID | B = Base of copper regen tube |
| | C = "B" regen, balls in ducts | C = 10mm Pb shot, 50mm scrn | C = 2d stage freezer tube near regen |
| | D = Small bore regen, empty | D = .5 m Cu tube, 1.6 mm ID | D = 2d stage freezer tube near cylinder |
| | E = "A" regen, balls in ducts | E = 30mm Pb shot, 30mm scrn | E = Warm end, 2d stg regen |
| | F = small regen, ducts open | F = 30mm ss bb, 30mm scrn | F = Cold end, 2d stg regen |
| | G = new plug, bypass tubes | G = 1.6 & 3.2mm ss bb | G = Expansion end, 2d stg cyl |
| | | H = G10 plug + 100 scrns | H = Base, 2d stg cylinder |
| | | | I = 1st stg cylinder head |

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